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LOW ENERGY CONSUMPTION HYDRAULIC TECHNIQUES



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19. ABSTRACT (Continue on reverse if necessary and identify by block number) This report documents the efforts to develop and test low energy consumption hydraulic techniques. Included is a discussion of the efforts to select and analyze a baseline hydraulic system (Phase I) and conduct trade studies on various candidate design approaches which could reduce weight and power consumption (Phase II). In addition, the report discusses the development of a test plan (Phase III) and the results of an endurance test (Phase IV). This program was directed at 8000 psi hydraulic system technology using a nonflammable hydraulic fluid chlorotrifluoroethylene (CTFE) although the concepts developed could be applied to any fluid at any system pressure. The concepts selected for test were variable system pressure, flow augmentation with load recovery valves and overlapped main valves. These concepts were embodied and tested with					
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18. Continued

Nonflammable Hydraulic Fluid
Load Recovery Valves
Pressure Intensification
Dry Sump Pumps
Variable Displacement Hydraulic Motors
Overlap Valves

19. Continued

a servocontrolled hydraulic pump and a dual tandem flight control servocylinder. The concepts were successfully tested and showed that significant weight and energy savings could be achieved on advanced aircraft.



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FOREWORD

This report was prepared by the McDonnell Aircraft Company (MCAIR) for the United States Air Force under contract number F33615-84-C-2417, which was conducted between August 1984 and April 1988. This contract was accomplished under Project Number 31453038. The work was administered under the direction of the Aero Propulsion Laboratory at the Air Force Wright Aeronautical Laboratories, Air Force Systems Command, Wright-Patterson AFB, Ohio. Air Force Program Managers were in turn Mr. K.E. Binns, Mr. A. G. Whitney and Mr. W. B. Campbell (AFWAL/POOS). Technical assistance with the hydraulic fluid was provided by Mr. C. E. Snyder and Mrs. L. Gschwender of the Materials Laboratory at Wright-Patterson AFB (AFWAL/MLBT). Mr. N. J. Pierce and Mr. J. B. Greene were Program Managers in turn, and Mr. R. E. Young and Mr. J. A. Wieldt were Principal Investigators in turn for MCAIR.

Phase I established a baseline aircraft hydraulic system based on the F-15 STOL Manuevering Technolgy Demonstrator (SMTD) Aircraft. The configuration of that aircraft's hydraulic power and flight control system was baselined as a combat survivable 8,000 psi hydraulic system using nonflammable CTFE (chlorotrifluoroethylene) hydraulic fluid. The systems were weighed out for comparison in studies of power efficient technologies.

Phase II applied various design approaches which could effectively reduce engine power consumption and weight in the hydraulic system. Life cycle cost comparisons were affected and the more effective concepts were selected for hardware development and tested in later program phases.

In Phase III, a demonstration hydraulic system and a test plan were defined to evaluate the concepts selected which were variable pressure, overlapped main control valves, flow augmentation and load recovery valves.

Phase IV of the program developed a hydraulic actuator which embodied three of the concepts. The Control Systems Division of the Parker Berteau Aerospace Group was contracted to build this actuation system. A pump which could provide variable supply pressure was developed by the Abex Corporation. Phase IV also included performance and endurance testing of a system that included these components.

This report encompasses the four phases listed above. During the contract period of performance, progress by program phase has been reported to the Air Force in two interim reports. These interim reports and the results of the last phase are compiled in this final report. To provide continuity and to improve readability, several figures are repeated within the document to avoid page referencing.

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LIST OF ABBREVIATIONS AND ACRONYMS

ABDR	Aircraft Battle Damage Repair
ACCM	Advanced Concept Cost Model
ACL	Analytical Chemistry Laboratory
ACM	Aircraft Combat Mission
ADP	Advanced Development Program
AFWAL/MLBT	Air Force Wright Aeronautical Labs/Materials Laboratory
AMAD	Airframe Mounted Accessory Drive
APC	Aircraft Power Controller
APM	Aircraft Porous Media Inc. (Pall Corporation)
CAS	Control Augmentation System
CASEE	Comprehensive Aircraft Support Effectiveness Evaluation
CDRL	Contractual Data Requirements List
cis	cubic inches per second
CLEAR	Component Life Evaluation and Reliability
CRES	Corrosion Resistant
CTFE	Chlorotrifluoroethylene
DDV	Direct Drive Valve
ECS	Environmental Control System
EHV	Electrohydraulic Valve
FA/LRV	Flow Augmentation/Load Recovery Valves
FAST	Flow Augmented Servovalve Technology
FCC	Flight Control Computer
FWFRHS	Flight Worthiness of Fire Resistant Hydraulic Systems
gpm	gallons per minute

LIST OF ABBREVIATIONS AND ACRONYMS (Continued)

HIM	Hydraulic Integrity Monitor
hp	horsepower
HSFR	Hydraulic System Frequency Response
HX	Heat Exchanger
HYTRAN	Hydraulic Transient Analysis
ILS	Integrated Logistics Support
IRAD	Independent Research and Development
LCC	Life Cycle Cost
LCOM	Logic Control Output Module
L/H	Left Hand
LECHT	Low Energy Consumption Hydraulic Techniques
LHS	Lightweight Hydraulic Systems
LORA	Level of Repair Analysis
LRV	Load Recovery Valve
LSA	Logistic Support Analysis
LTD	Laboratory Technology Demonstrator
LVDT	Linear Variable Displacement Transducer
LVR	Local Velocity Reduction
MBC	Metal Bellows Corporation
MCAIR	McDonnell Aircraft Company
MCV	Main Control Valve
MFHBF	Mean-Flight-Hours-Between-Failure
MMH/FH	Maintenance-Man-Hours-Per-Flight-Hour
MTBMA	Mean-Time-Between-Maintenance-Action
MTBF	Mean-Time-Between-Failures
MTTR	Mean-Time-To-Repair

LIST OF ABBREVIATIONS AND ACRONYMS (Continued)

NDI	Nondestructive Inspection
NHPSTA	Nonflammable Hydraulic Power System for Tactical Aircraft
OASHA	Operating and Support Hazard Analysis
OPS	Operational and Support
ORLA	Optimum Repair Level Analysis
PC(1 & 2)	Primary Control System
PHA	Preliminary Hazard Analysis
psi	pounds per square inch (lbs/in ²)
psid	pounds per square inch (lbs/in ²) differential
psig	pounds per square inch (lbs/in ²) gage
RDT&E	Research, Development, Test Evaluation
R/H	Right Hand
R&M	Reliability and Maintainability
RHI	Real Hazard Index
rpm	revolutions per minute
SMTD	STOL Maneuvering Technology Demonstrator
SOW	Statement of Work
SSFAN	Steady State Flow Analysis
STOL	Short Takeoff and Landing
UT(1 & 2)	Utility System
VDHM	Variable Displacement Hydraulic Motor
WPAFB	Wright-Patterson Air Force Base

SECTION I INTRODUCTION AND SUMMARY

1.1 INTRODUCTION - Air Force Contract F33615-84-C-2417, "Low Energy Consumption Hydraulic Techniques (LECHT)," was awarded to McDonnell Aircraft Company (MCAIR) effective 24 August 1984. This report covers all four phases of the program, including Advanced Aircraft Hydraulic System Selection (Phase I), Tradeoff Studies (Phase II), Detail Design (Phase III), and Component Fabrication and Demonstration Tests (Phase IV).

1.1.1 Background - A state of the art hydraulic system design can still result in low system efficiency and poor energy utilization at most flight conditions. Present systems are sized for peak loads, low temperature operation, and maximum actuator hinge moment and rates. These conditions may not occur simultaneously. At the same time, the advent of relaxed static stability and increased maneuvering requirements for future aircraft will result in higher rates and more control surfaces. The hydraulic peak power will be 2 or 3 times higher for these new aircraft if conventional design techniques are used.

The purpose of this program was to investigate energy management techniques which could result in optimum utilization of hydraulic system power. This program showed the potential for providing significant weight savings and lower heat rejection for future aircraft hydraulic systems.

1.1.2 Program Objective - The objective for this effort was to analyze, design, develop, and demonstrate energy management techniques for reducing the power and size of aircraft hydraulic systems.

1.1.3 Program Plan - The program was broken down into four phases. Phase I included aircraft selection and the establishment of baselines for the hydraulic system characteristics and Life Cycle Cost (LCC). Phase II evaluated and selected low energy consumption candidates through trade studies using the evaluation criteria developed in Phase I. Phase III developed detailed designs of components using the low energy consumption concepts selected in Phase II. In Phase IV, the components were fabricated, then tested in a demonstration system.

1.2 SUMMARY

1.2.1 Phase I - Phase I included aircraft selection and establishment of baselines for the hydraulic system characteristics and LCC. Evaluation criteria for Phase II trade studies were established and several concepts were identified that showed promise for saving hydraulic energy.

- o Intelligent Pumps
- o Overlapped Valves
- o Flow Augmentation/Load Recovery Valves
- o Pressure Intensifiers
- o Flow Augmented Cooling
- o Dry Sump System Pumps/Motors
- o Servohydraulic Motors
- o Variable Usage/Displacement Actuators
- o Optimized System/Subsystem Applications

a. Aircraft Selection - The F-15 STOL Maneuvering and Technology Demonstrator (SMTD) aircraft was selected to meet program requirements.

b. Hydraulic System Baseline - The approach was to establish the aircraft baseline hydraulic systems with Chlorotrifluoroethylene (CTFE) fluid at 8,000 psi, using direct drive valves (DDV) where feasible. Baseline hydraulic parameters established were:

- o Total Hydraulic System Weight
- o Heat Exchanger Weight
- o Reliability
- o Maintainability
- o Life Cycle Cost
 - Hydraulic Subsystem Level
 - Total Aircraft

c. Evaluation Criteria for Low Energy Consumption Concepts - Evaluation of concepts was as shown in Figure 1. Each concept was incorporated into a component and the components were evaluated for the total hydraulic system effects on:

- o Hydraulic System Weight
- o Hydraulic System LCC
- o Hydraulic System Heat Load
- o Total Aircraft Weight
- o Total Aircraft LCC

Weighing factors for the parameters were established during the initial Phase II trade studies.

1.2.2 Phase II - The objectives of the second phase were to:

- o Evaluate low energy concepts using criteria developed in Phase I
- o Identify energy savings concepts and evaluate in trade studies
- o Establish and design a laboratory test system

Four energy reduction concepts were selected for trade studies:

- o Variable pressure (Intelligent Pump) pump
- o Overlap valve
- o Flow augmentation with load recovery valves
- o Dry sump pump

Variable displacement motors, variable area actuators and pressure intensifiers were also considered. The variable area actuator was not selected because it increased complexity and was considered redundant to the use of the variable pressure (intelligent pump). Analysis of both the variable displacement motor and pressure intensifier indicated considerable potential. However, their development to hardware and verification of potential has not been accomplished as compared to the selected concepts.

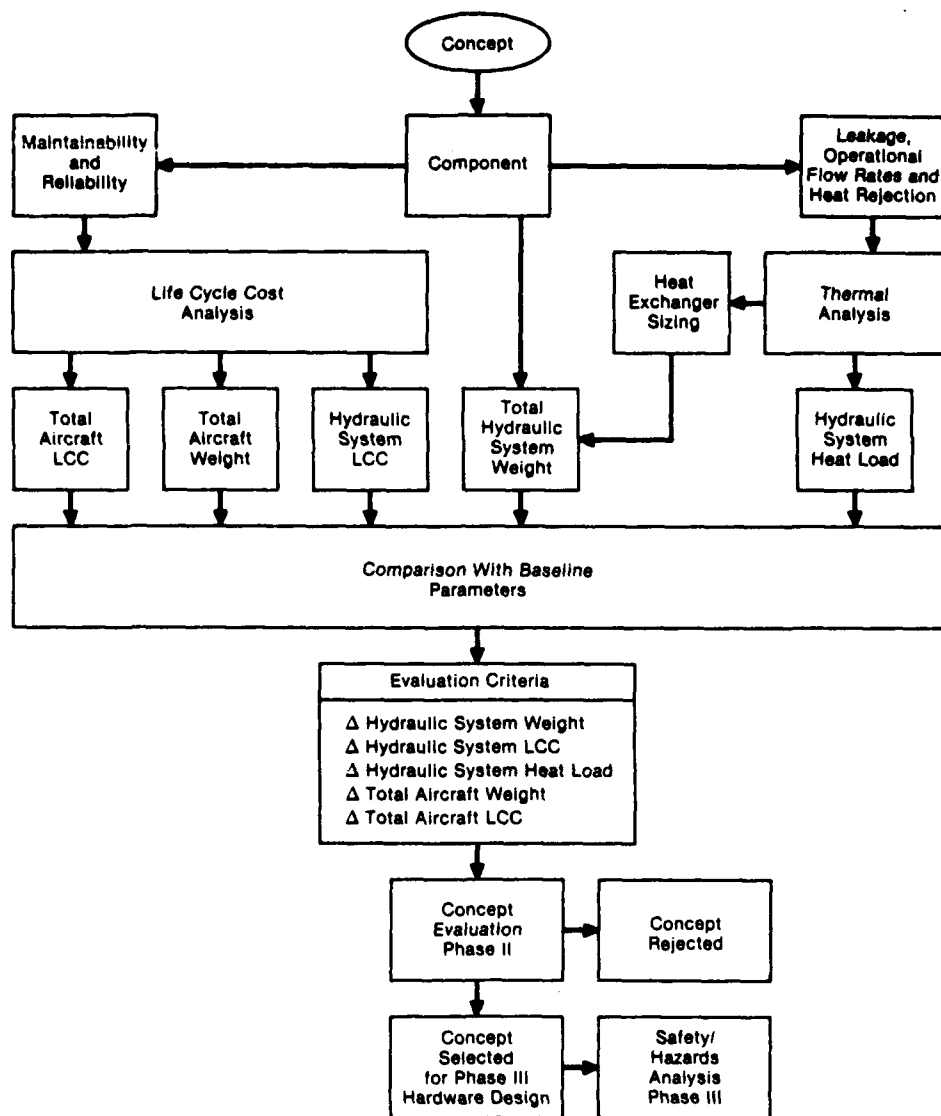


Figure 1
Phase II Evaluation Criteria

a. Summary of Concept and Baseline Evaluations - Figure 2 summarizes the results for the five concepts and three baselines that were evaluated. The baseline configurations used the F-15 SMTD Hydraulic System layout with various pressures and fluids. The heat rejection of the hydraulic system and the pump was determined for input into the thermal analysis. A ram air heat exchanger was then sized to limit hydraulic system fluid temperature to 275°F maximum. The hydraulic system weight was calculated, along with the associated changes in the weight of components and distribution systems. Total aircraft weight was calculated using the LCC model.

Parameter	3,000 psi		8,000 psi					
	Baseline MIL-H-83282	Baseline CTFE	Baseline CTFE	Overlap Valve	Dry Sump	Flow Aug	Intelligent Pump (3,000/ 8,000)	Combination (3,000/ 8,000)
Hydraulic System Heat Rejection - HP	64.7	64.7	99.2	61.0	85.0	84.0	40.3/99.2	17.8/35.4
Ram Air Heat Exchanger Weight - lb	85.7	85.7	192.6	139.3	184.7	182.0	77.6	25.8
Hydraulic System Weight - lb	1,982.99	2,224.11	1,734.03	1,680.73	1,718.31	1,700.84	1,645.03	1,551.80
Total Aircraft Weight - lb	28,560.0	29,909.0	28,047.0	27,928.0	28,012.0	27,973.0	27,849.0	27,641.0
Δ Baseline Hydraulic System Unit Flyaway Cost - \$M	Reference for Baseline	+0.004	+0.14					
Δ Baseline Aircraft Unit Flyaway Cost - \$M	Reference for Baseline	+0.004	+0.088					
Δ Baseline Hydraulic System Life Cycle Cost - \$M	Reference for Baseline	+2.0	+96.0					
Δ Baseline Aircraft Life Cycle Cost - \$M	Reference for Baseline	+203.0	-112.0					
Δ 8,000 psi Hydraulic Concept Life Cycle Cost - \$M			Reference for Baseline	-13.0	+22.0	+2.0	+7.0	-29.0
Δ Aircraft With 8,000 psi Concept Life Cycle Cost - \$M			Reference	-104.0	-4.0	-53.0	-142.0	-337.0

Figure 2
Tradeoff Summary

Results showed that the 8,000 psi CTFE baseline hydraulic system costs 140 thousand dollars more per aircraft than the baseline 3,000 psi MIL-H-83282 system. However, because the total aircraft weight was less, the CTFE system showed a 112 million dollar LCC savings for 500 aircraft.

Compared to the 8,000 psi CTFE baseline, the variable pressure (intelligent pump) showed a total LCC savings of 142 million dollars. Overlap valves saved 104 million dollars, followed by flow augmentation at 53 million dollars and the dry sump pump at 4 million dollars. The combination of all four concepts showed a total LCC savings of 337 million dollars.

The dry sump pump LCC benefits were relatively small and because insufficient development had been accomplished to ensure operational practicality, it was eliminated from further consideration. The concepts selected for Phase III development were:

- o Overlap valves
- o Flow augmentation
- o Load recovery valves
- o Intelligent pump

b. Laboratory Test System - The proposed laboratory test system for Phase I design and Phase IV testing is shown in Figure 3. The test setup formerly used for the 750 hour demonstration in the Flight Worthiness of Fire Resistant Hydraulic Systems (FWFRHS) (Reference 1) program was used.

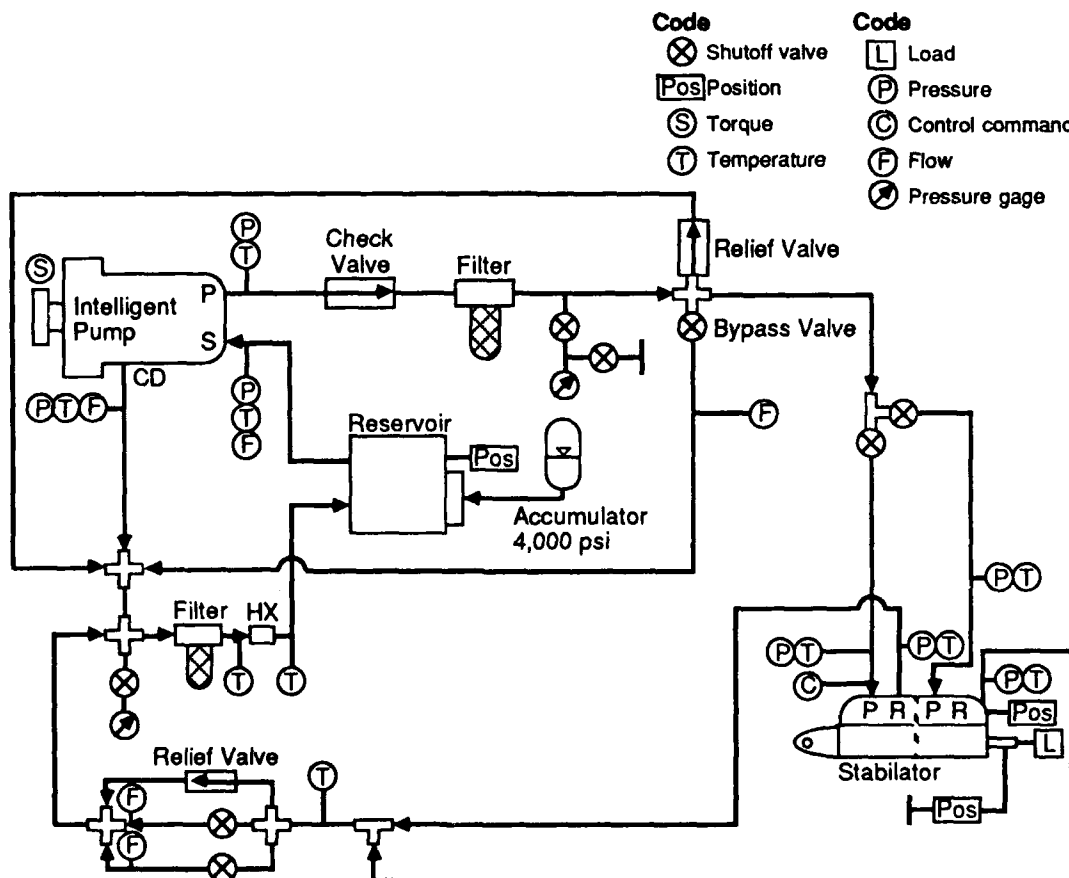


Figure 3
LECHT Demonstration System Schematic

Instrumentation was provided for flows, pressures, temperatures and position, as shown in Figure 3. Simulated aircraft loads were provided on the stabilator servoactuator. Three-micron filtration was provided on the high pressure side and five-micron filtration on the return side. A heat exchanger was used to stabilize the fluid at the desired test temperature.

The tests included measurement of input torque (inch-pounds) to the pump over a specified time interval to measure energy (horsepower) consumed with and without each energy savings concept. The difference between the horsepower values with and without the concept was the energy savings. In addition, other appropriate tests were conducted such as overlap valve null leakage vs. line-to-line valve null leakage.

Overlap Valve - Figure 4 illustrates the concept of overlapped valves, which reduces hydraulic power requirements by reducing null leakage across the servovalve. The overlap servovalve installed in the stabilator valve manifold was tested and compared to the conventional line-to-line servovalve. A servoactuator cycling sequence was established and energy measurements were taken.

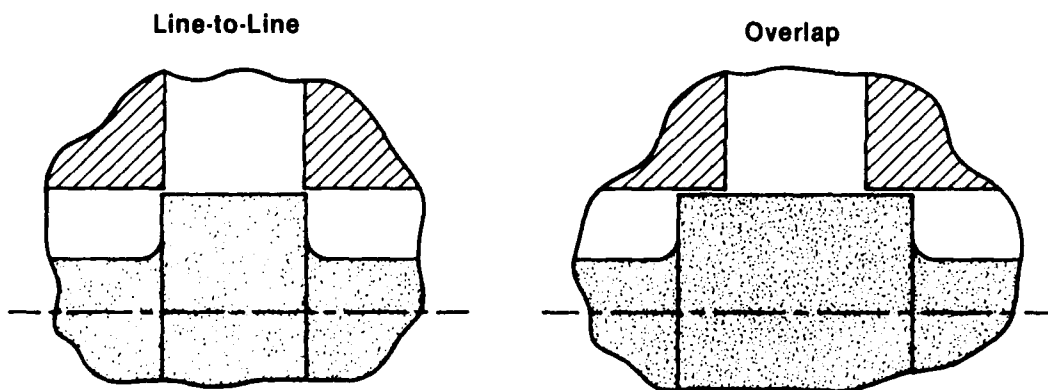


Figure 4
Overlap Valve Concept

Load Recovery Valves - Figure 5 shows a schematic of the load recovery valves, which are check valves that allow fluid to short circuit from the return side of the piston to the pressure side when an assisting load causes the pressure side to drop below the actuator outlet pressure. Testing was accomplished with load recovery valves installed in the stabilator valve manifold to allow energy comparisons with the baseline system. The cross hatched area shows the increased performance attained with load recovery valves.

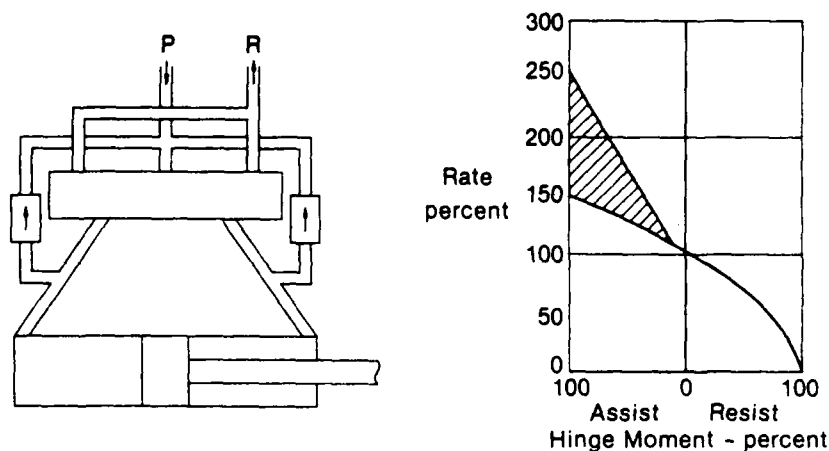


Figure 5
Load Recovery Valves

Flow Augmentation - Flow augmentation uses a jet pump to recirculate return flow to the pressure side when low load operation occurs (see Figure 6). The reduced pump flow demand allows a smaller pump and supply lines, which translates into reduced system heat rejection and lighter system weight. Tests were performed with and without the flow augmentation devices installed in the stabilator valve manifold. Conditions of varying loads were tested in the actuator cycling sequence. Identical sequences were run for the baseline and with flow augmentation. Figure 7 presents the combined performance impact of flow augmentation and load recovery valves (LRV). The LRVs prevent jet pump cavitation during the assisting portion of the actuator stroke.

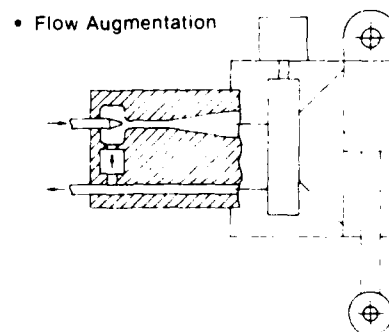
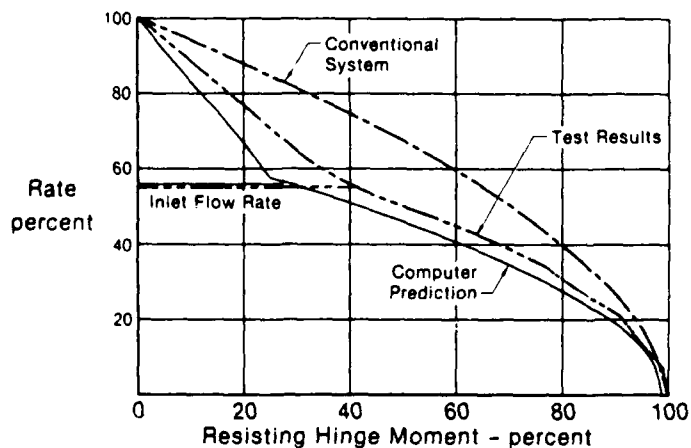


Figure 6
Flow Augmentation

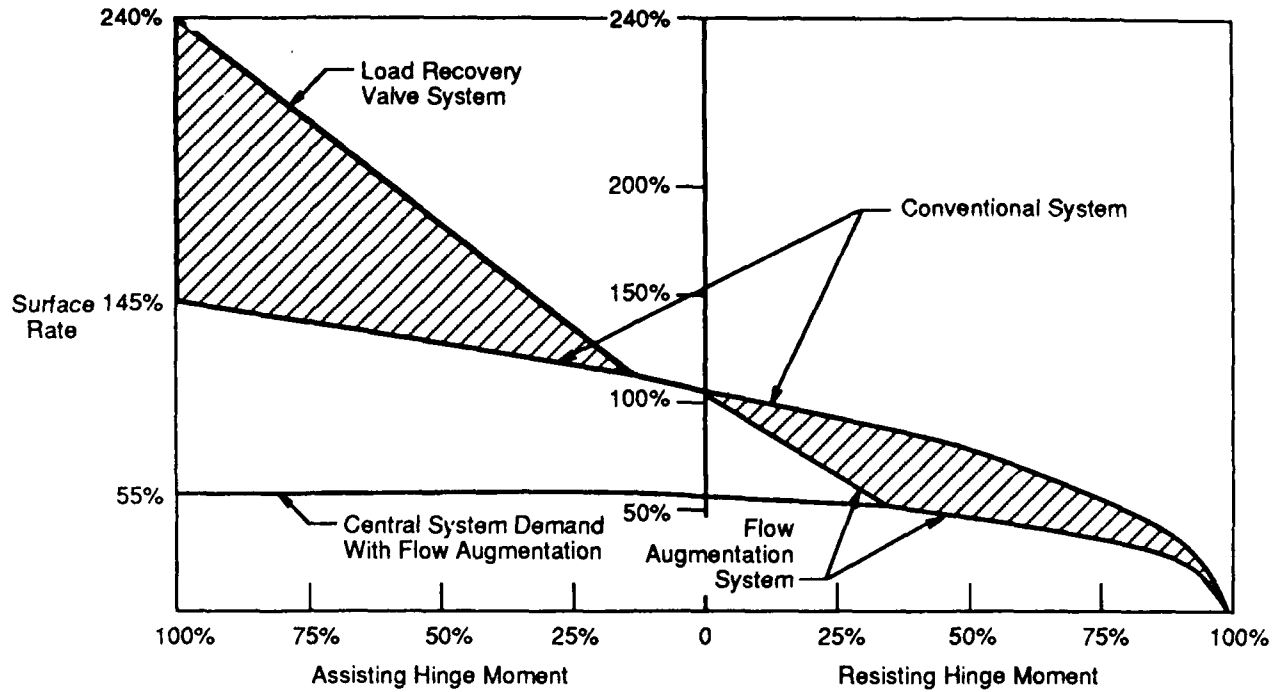


Figure 7
Combined Jet Pump/Load Recovery Valve Performance

Variable pressure (intelligent) pump - During periods of low flight control actuator flow demand, intelligent pumps (shown in Figure 8) can be operated at a lower pressure setting to reduce energy consumption and reduce heat rejection. To establish baseline performance values for energy

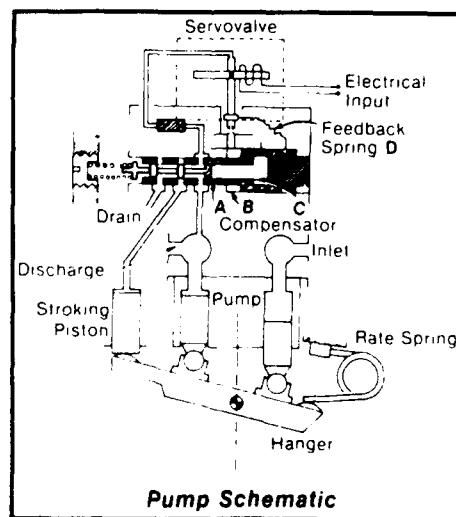


Figure 8
Intelligent Pump

consumption, the electrohydraulic valve (EHV) was commanded to drive the pump compensator position to a fixed 8,000 psi outlet pressure. In the variable pressure mode, the compensator was controlled by the EHV. An actuator cycling sequence was established to provide a system load. Identical sequences were then performed for both pump configurations.

1.2.3 Phase III - The third program phase was the detail design of the test setup, procurement of hardware, creation of a test plan and the PHA.

a. Design and Specification Definition - A test system was designed to demonstrate and evaluate the four candidate concepts listed below:

- o Variable pressure (intelligent) pump
- o Overlapped valve
- o Flow augmentation
- o Load recovery valves

The demonstration system for the FWFRHS program was modified to include a variable pressure pump, a loaded stabilator actuator with provisions for an overlap valve, flow augmentation devices, and load recovery valves, and is shown in Figure 3.

The 15 gpm pump, which can be varied between 3,000 and 8,000 psi outlet pressure or can be run at a fixed pressure setting, was controlled by monitoring the main control valve Linear Variable Displacement Transducer (LVDT) position signal. This signal was multiplied by the flow gain to get an estimate for flow demand, which was used to schedule pump pressure. The pump is an open loop design, manufactured by the Abex Corporation for use in MCAIR's 8,000 psi efforts, and uses an EHV to control pump pressure.

The 8,000 psi, CTFE version of the F-15 stabilator servoactuator was designed and fabricated by Parker Berteau. It is a dual tandem actuator with the same force output, stroke, and mounting attachments required for the F-15. Two main control valve (MCV) spool and sleeve assemblies were supplied, one line-to-line and the other with a 0.003-inch overlap. In addition, provisions for the installation of load recovery valves and flow augmentation devices were made. The load recovery valves were supplied along with the actuator, while the jet pumps were manufactured by the LEE Company. Crissair Inc., designed a special -10 (5/8 in.) Rosan fitting incorporating a check valve and filter screen.

The lines, filters, and reservoir represent an aircraft installation. Aircraft Porous Media (APM) provided a three-micron pressure filter and a five-micron return filter. The reservoir was a Metal Bellows Corporation (MBC) 8,000 psi bootstrap type, with a 4,000 psi accumulator providing constant pressure. Appropriate check valves, relief valves, an accumulator and a heat exchanger were also included.

b. Test Plans - Test plans for evaluation of the various concepts were as follows:

- o Pump acceptance testing, including leakage and heat rejection tests, hysteresis and linearity, and frequency response.
- o Baseline actuator testing, including null leakage, hysteresis, valve reversals, loaded step responses, and frequency response.
- o Individual concept testing which included pertinent tests performed with overlap valves, load recovery valves, jet pumps and variable pressure installed sequentially.
- o System performance included testing to measure system energy consumption with all concepts in place.
- o 200-hour endurance test.

c. Preliminary Hazard Analysis - A Preliminary Hazard Analysis (PHA) was performed in accordance with MIL-STD-882A to determine if any of the concepts would present hazardous conditions in an aircraft application or as they are incorporated in the demonstration test. According to the results of the PHA, all identified hazards were an acceptable risk.

1.2.4 Phase IV - The fourth phase of the program included for the fabrication of the components and the demonstration system, followed by the testing of the components and concepts.

a. Component/System Fabrication - The variable pressure pump was installed in the test setup, and a controller box was designed and built at MCAIR. The stabilator actuator was mounted into a test fixture designed to simulate aircraft geometry, inertia, stiffness and load conditions.

b. Component/System Testing - Acceptance tests were performed on the 15 gpm pump with no major problems detected with the unit. It was noticed however, that case drain flow often increased with decreased pressure setting. This was identified as being a function of the EHV control requirements and return flow, which was dumped into the pump case. Baseline actuator testing, which included the line-to-line valve, plugs replacing the load recovery valves, and no jet pumps, established a maximum no-load rate of 8.5 in./sec in each direction. Frequency response tests at 8,000 psi showed performance within the F-15 SMTD specification values.

Individual concept testing was performed to evaluate and quantify the benefits associated with each concept.

(1) Overlap Valve Concept - The primary benefit of the 0.003-inch overlap valve was in the reduction of null leakage, which in turn reduced steady-state energy consumption. Figure 9 shows a comparison of null leakages for the line-to-line valve, and two different 0.003-inch overlapped spools.

Valve Configuration	Fluid Temperature (°F)	System Pressure (psi)		
		8,000	5,500	3,000
Line-to-Line	160	0.84	0.70	0.50
	275	0.85	0.77	0.52
0.003 in. Overlap No. 1	160	0.18	0.20	0.18
	275	N/A	N/A	N/A
0.003 in. Overlap No. 2	160	0.32	0.39	0.33
	275	0.44	0.50	0.34

Figure 9
Comparison of Actuator Null Leakages
With Various MCV Configurations

At 8,000 psi system pressure and 50 psi on the return side, a 62 to 78 percent reduction in null leakage and horsepower was achieved. The results at 3,000 psi were somewhat less at 34 to 64 percent reduction.

An expected degradation of performance during ± 1 percent amplitude frequency response tests was noted, as well as an order of magnitude change in threshold. However, previous F-15 iron bird and F-18 simulator evaluations showed that aircraft handling qualities were acceptable with overlap valves. In fact, during target tracking maneuvering, handling qualities with overlap valves were superior to line-to-line valves. At larger amplitudes, overlap and line-to-line valve performance were very similar.

(2) Flow Augmentation - As shown in Figure 10, the jet pump allowed a 32 to 56 percent reduction in central system flow demand for a comparable no load rate. In an aircraft application, this concept would allow a significant reduction in pump size, and therefore a corresponding decrease in heat rejection or wasted energy. It is important to note that the flow augmentation and load recovery valves together, with a full assisting load applied, allowed a 49 to 63 percent reduction in central system flow demand.

(3) Load Recovery Valve - As shown in Figure 11, the performance of the actuator with load recovery valves in the extend direction of operation is quite close to predicted performance. The rate increase for the load recovery valve system as compared to the conventional system, was approximately 2.85 times in the 90 to 100 percent assisting load range. On initial testing of the actuator, the LRV function was found to be inoperative. The actuator was returned to Parker Bertea where it was determined that the LRV function was impaired due to improper machining of the poppets, resulting in the valves sticking. New poppets were installed and the unit was shipped back to MCAIR for further testing. During testing, the unit performed as expected in the extend direction, but the LRVs did not operate properly in the retract direction. To verify that it was not another poppet failure, MCAIR switched

No Load Comparison	Conventional System		Flow Augmentation System		Percent Flow Reduction
	Flow (GPM)	Rate (in./sec)	Flow (GPM)	Rate (in./sec)	
Extend Direction	9.1	8.5	4.0	8.3	56%
Retract Direction	9.5	8.5	6.5	8.5	32%
Average Reduction					44%

Full Assisting Load Comparison	Conventional System		Flow Augmentation System		Percent Flow Reduction
	Flow (GPM)	Rate (in./sec)	Flow (GPM)	Rate (in./sec)	
Extend Direction	13.5	10.5	5.0	30.0	63%
Retract Direction	14.7	11.0	7.5	12.5	49%
Average Reduction					56%

Figure 10
Conventional vs. Flow Augmentation

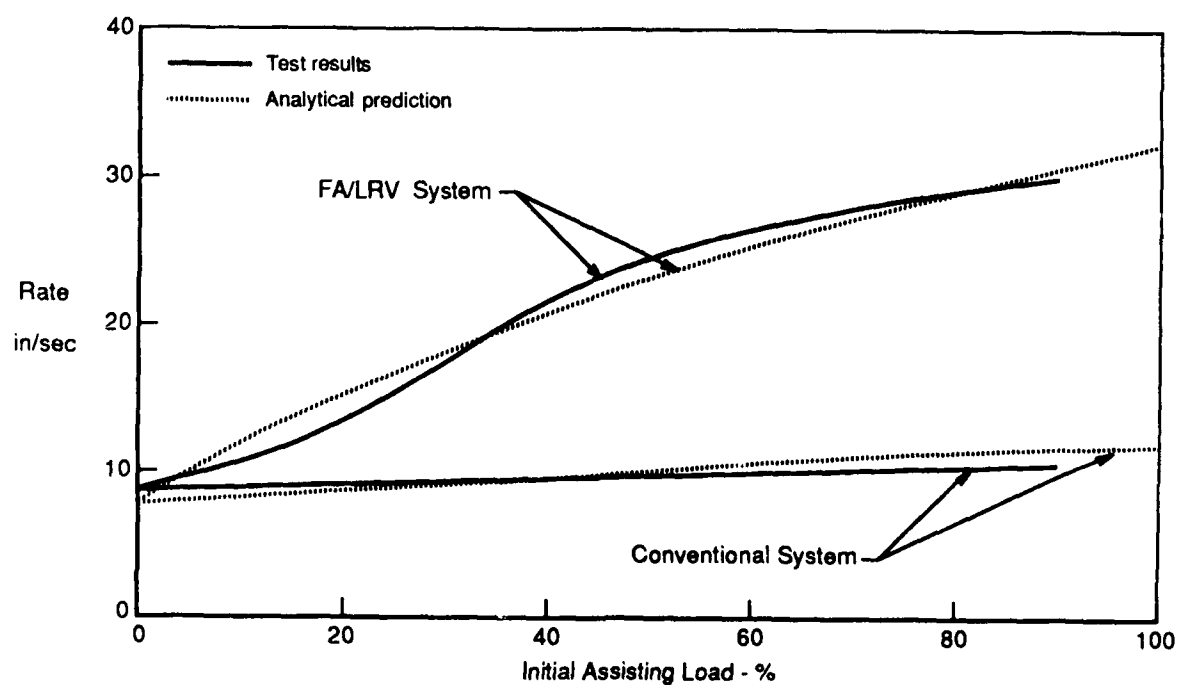


Figure 11
Actuator Extend Direction, Assisting Load Performance Comparison

the LRVs. However, this had no effect on the performance of the unit. All indications were that a port was not fully machined.

(4) Variable Pressure - The benefit associated with variable pressure is the reduction in energy consumption and heat rejection during periods of low surface activity. Pump torque was recorded during speed sweeps between 1,500 and 4,500 rpm at 3,000, 5,500, and 8,000 psi (Figure 12), and revealed a 45

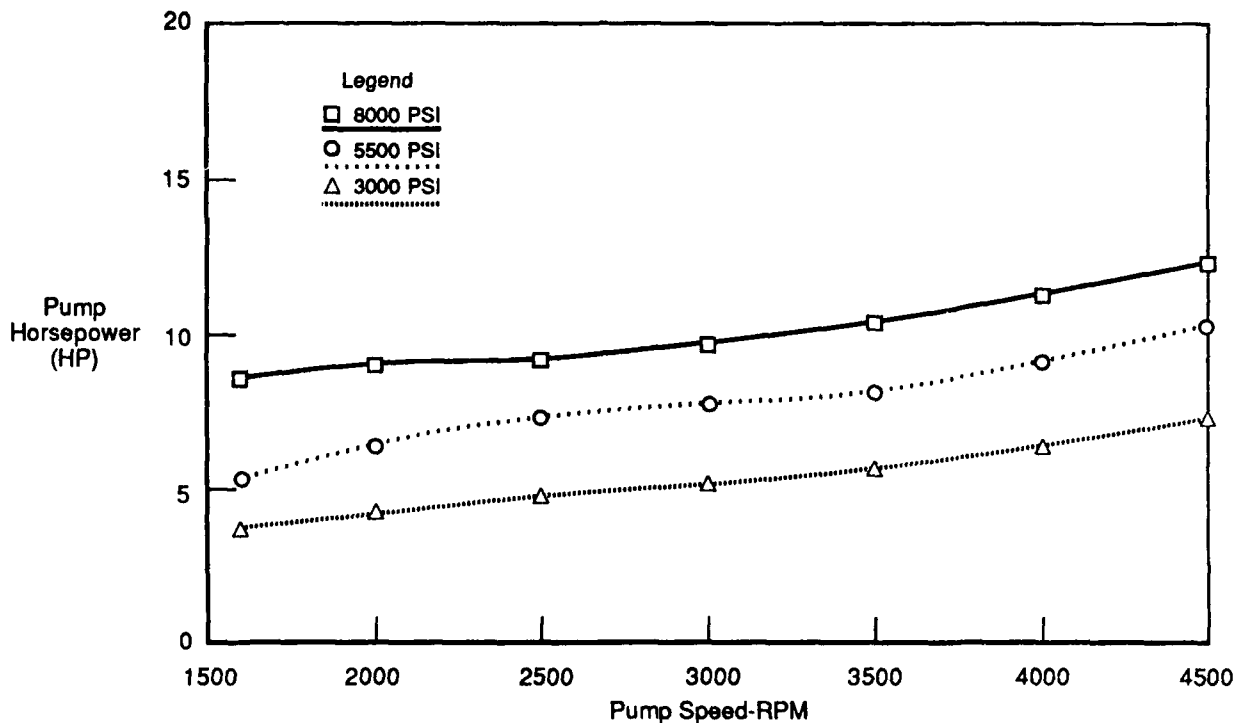


Figure 12
Pump Heat Rejection at 160°F Pump Inlet Temperature

percent reduction in pump steady-state horsepower when operating at 3,000 psi vs. 8,000 psi (6.1 vs. 11.1 horsepower). This data was further verified during the endurance test, as shown in Figure 13. Pump torque was recorded during a 72-second duty cycle consisting of 75 percent small amplitude cycles (less than 10 percent main ram amplitude) and 25 percent large amplitude cycles. The pump torque was then integrated to get a measure of total energy. Results indicated that the variable pressure pump system operation required 48 percent less energy than constant 8,000 psi operation.

(5) 200-Hour Endurance Test - Only 139.4 hours of the scheduled 200-hour endurance test was completed on the demonstration system because:

- o Problems developed during the endurance test and supplier spare parts were not available under the original contract to guarantee the needed repairs. These problems are further discussed in this section.

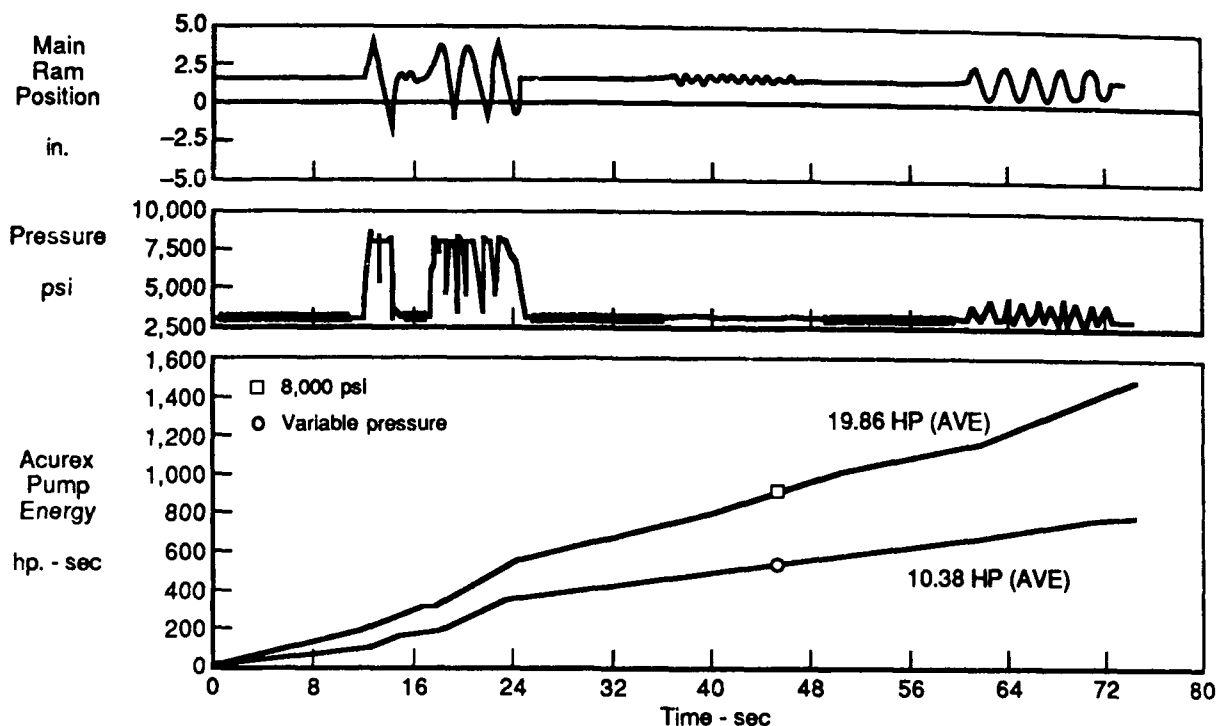


Figure 13
Total Pump Energy Consumption Over 72-Second Duty Cycle

- o MCAIR and the Air Force were in concurrence to stop testing in order to conclude the program without further contract extensions.

The following is a list of components with total hours accumulated:

Variable Pressure Pump (S/N 193174)	245.0 total hours
Variable Pressure Pump (S/N 193175)	47.9 total hours
Servoactuator	260 total hours (estimated)

The servoactuator completed 697,000 endurance test cycles plus at least another 100,000 performance test cycles.

Several problem areas were discovered with the variable pressure (intelligent) pump and the actuator, the more critical of which are listed below:

- o Pump Electrohydraulic Valve (EHV) - Four separate failures occurred to the pump EHV, at least one of which appeared to have been caused by excessive contamination. The failure of the return filter was a major contributing factor. One of the failures was due to a bad O-ring on a control sleeve, which caused the pump low pressure setting to vary with fluid temperature.

- o Servoactuator Transfer Tube - Three failures of the transfer tube, including a beefed up unit, revealed an inadequate column loading design. The design used was a straight and bent tube attached at only one end. High differential pressures caused the tube to buckle.
- o Servoactuator Main Control Valve LVDT - A production F/A-18 3,000 psi LVDT was used for Main Control Valve (MCV) position feedback, and was proved to be totally inadequate. The LVDT was exposed to return system pressure spikes of approximately 4,500 psi (which are considered normal for an 8,000 psi system), resulting in numerous fatigue related failures, usually at welds.

SECTION II
PHASE I - ADVANCED AIRCRAFT HYDRAULIC SYSTEM SELECTION

Phase I included Task 1 selection of the aircraft and definition of its hydraulic system, and Task 2 included definition of the evaluation criteria for new concepts. Also included were the efforts involved in determining the baseline configurations and identifying candidate concepts.

2.1 ADVANCED AIRCRAFT SELECTION

2.1.1 Task 1 - Select System and Define Baseline - The F-15 SMTD aircraft was selected as the advanced fighter for this study. It was chosen because:

- o It was an enhanced existing fighter aircraft that was updated to have a fly-by-wire control system with loads and rates associated with future, highly maneuverable aircraft.
- o The flight control actuators incorporated direct drive valves with multichannel electrical control inputs.
- o It employed digital flight control computers.
- o The hydraulic systems already had many of the requirements and test data documented. This provided a baseline for comparison and a foundation for more valid tradeoff studies.
- o Component system cost, reliability, and maintainability data were also available.

a. F-15 SMTD Hydraulic Systems - The basic production F-15 utilized three separate 3,000 psi hydraulic systems, see Figure 14. The systems were Type II, per MIL-H-5440, and were designed to operate with MIL-H-5606 fluid at

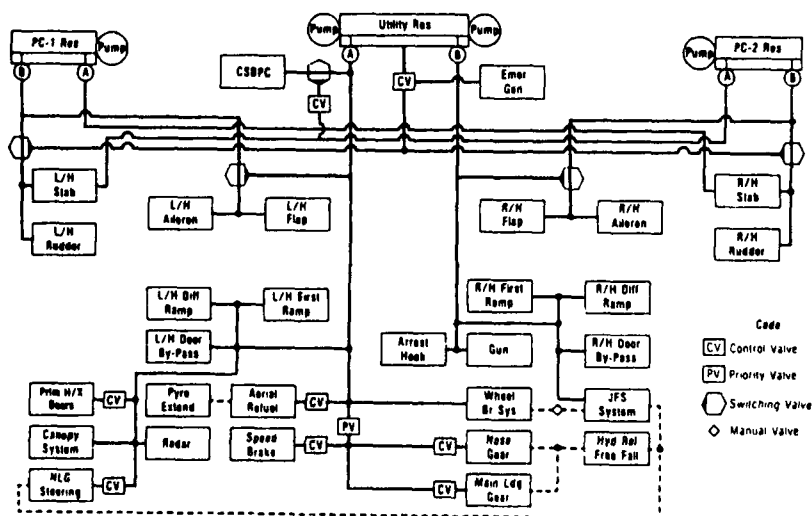


Figure 14
Production F-15 Hydraulic System

temperatures from -65°F to 275°F. When the MIL-H-83282 fluid requirement was introduced into the F-15, the minimum operating temperature limit was changed to -40°F.

Power Control Systems 1 and 2 (PC-1 and PC-2) were used to power the primary flight control actuators. The Utility System powers the various remaining subsystems and was automatically switched into the flight control servoactuators in the event of loss of either PC-1 or PC-2, unless the loss of the PC system was caused by a leak downstream of the switching valve.

Figure 15 shows the F-15 SMTD which added canard and engine nozzle control actuators to the hydraulic systems and changed the trailing edge flap actuators to variable position flaperon actuators. These changes increased the flow demands.

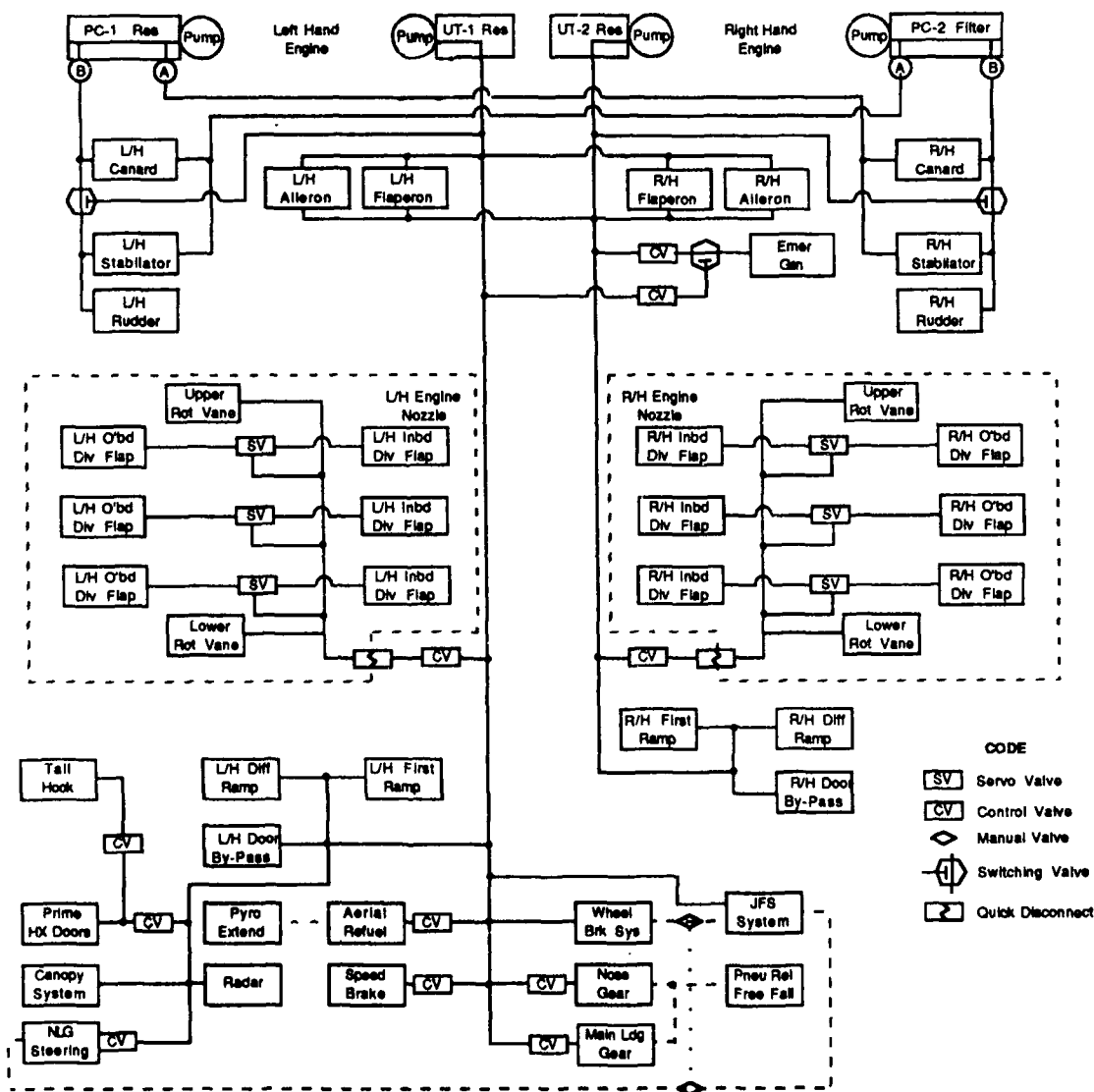


Figure 15
F-15 STOL and Maneuvering Technology Demonstrator (SMTD)
Hydraulic System

The engine nozzle control actuators were placed on the utility hydraulic system and the canard actuators were placed on the PC hydraulic systems. To ensure that a single failure in the two pump utility system did not cause a loss of hydraulic power to both sets of engine nozzle actuators simultaneously, the utility hydraulic system was divided into two separate systems (UT-1 and UT-2), with one pump for each system. The two utility systems provided power for the ailerons, flaperons and engine actuators for the engine nozzle control, vectoring and reversing. In addition, they provided power for the secondary control systems, i.e., landing gear, brakes, speedbrake, etc., the engine inlets, the emergency generator, and other utilities, as well as backing up the stabilator and rudder after loss of PC-1 or PC-2.

Each PC system was divided into two branches, with each branch protected by the Reservoir Level Sensing (RLS) valves. The emergency generator was operated off utility system UT-2, with utility system UT-1 as a backup through a switching valve. The switching valve and plumbing downstream to the emergency generator were isolated from both utility systems by normally open (power to close) solenoid valves upstream of the switching valve. These valves were operated simultaneously by the aircraft electrical system using the same logic for control of the emergency generator as the production aircraft. During normal operation after start up, the valves were closed unless a failure occurred.

When the fire extinguishing system was armed, each engine nozzle hydraulic system was isolated from the aircraft hydraulic system by electrically operated shutoff valves at the engine firewall. The engine nozzles were connected to the aircraft hydraulic systems using self-sealing quick disconnects in the engine compartment at the engine/airframe interface, to preclude contamination or air in either system upon removal of an engine.

Direct drive valves were provided on the flight control actuators.

b. LECHT Program CTFE 8,000 psi Baseline Hydraulic System - An 8,000 psi baseline hydraulic system was established for study comparison in Phase II. The F-15 SMTD hydraulic system was modified to a representative production aircraft configuration with one utility hydraulic system having two pumps.

Figure 16 shows the LECHT CTFE 8,000 psi baseline hydraulic system. The hydraulic pumps and distribution system were sized to provide flow rates under maximum rate and simultaneous subsystem operation. The actuator force outputs were sized to be the same at 8,000 psi as at 3,000 psi. All hydraulically actuated subsystems were designed to use direct drive valves where feasible.

The left-hand and right-hand engine nozzle actuators are normally powered by the utility hydraulic system. Switching the valves provide backup from the PC hydraulic systems.

2.1.2 Task 2 - Study Evaluation Criteria Definition - As noted in Section 1, the five evaluation criteria for low energy consumption concepts are the heat load, LCC, weight of the hydraulic system, and the weight and LCC of the aircraft. Heat load/heat reduction and the energy reduction of

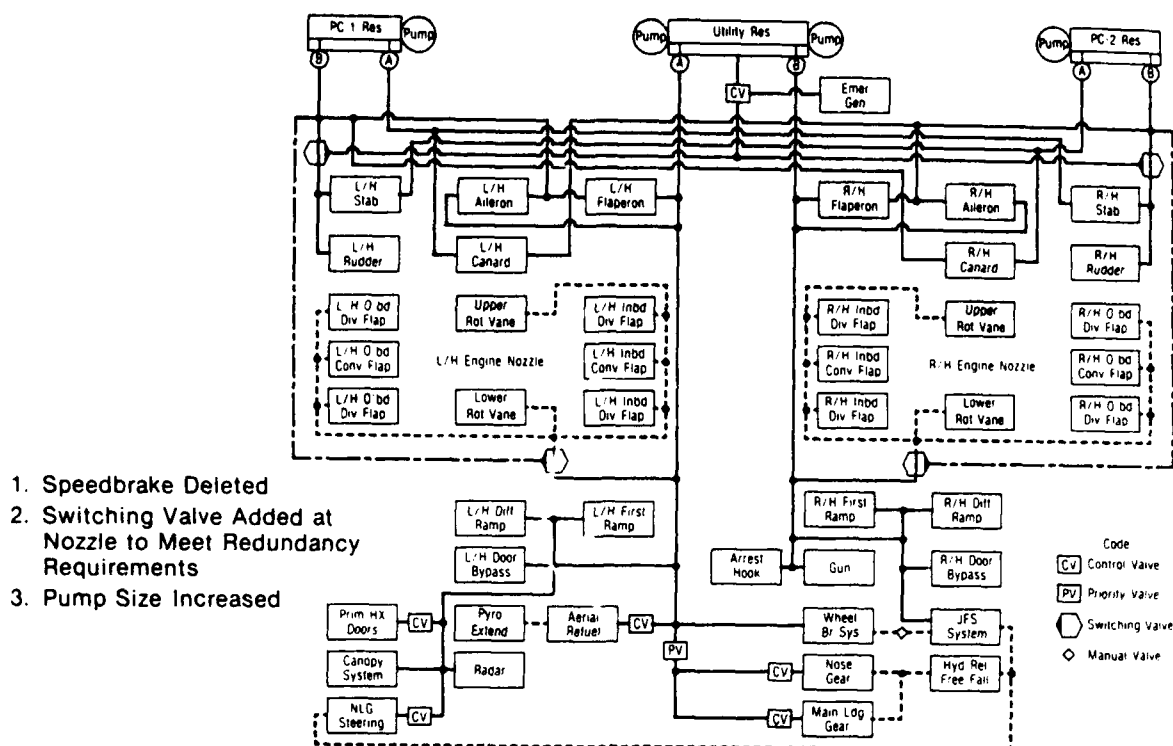


Figure 16
F-15 Production Configuration SMTD
 CTFE Fluid at 8,000 psi

each study concept were determined by the MCAIR Thermodynamics Heat Transfer Program, which also supplied models of the F-15 fuel/hydraulic system. A baseline thermal model with heat rejection and heat exchanger size was established.

LCC analyses also provided data for comparing energy savings concepts, and considered weight, complexity, reliability and maintainability. LCC analyses provided evaluation criteria for three parameters: (1) total aircraft weight, (2) hydraulic system LCC, and (3) total aircraft LCC.

The Steady-State Flow Analysis (SSFAN) computer program was used for resizing the F-15 SMTD hydraulic system to establish a baseline hydraulic system weight. This weight was input into the LCC analyses as well as being used as a baseline evaluation criterion for direct weight comparison.

a. Hydraulic System Weight - To evaluate the weight impact of the low energy hydraulic consumption concepts, a baseline weight was established for the F-15 SMTD 8,000 psi CTFE hydraulic system. The hydraulic system weight is categorized as shown in Figure 17. Component and distribution system design for 8,000 psi used revised design factors and the newest available materials. The following goals were established for the 8,000 psi baseline:

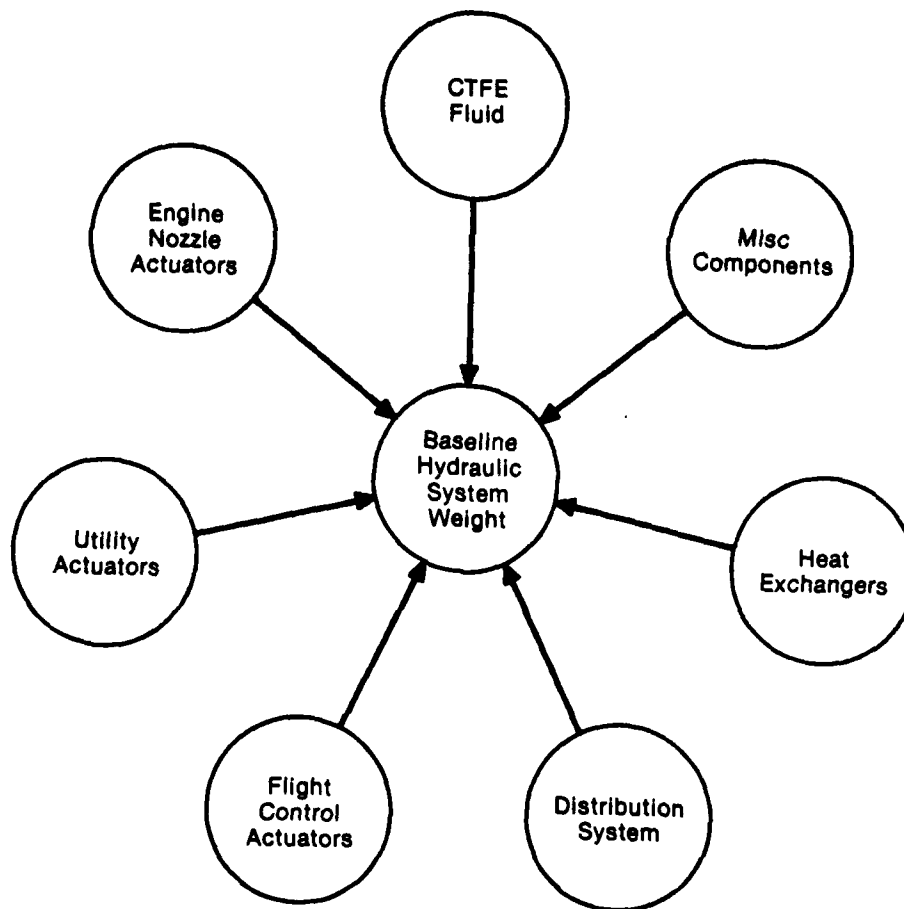


Figure 17
CTFE 8,000 psi F-15 SMTD Hydraulic System Weight

- o Re-evaluate design requirements
 - Fatigue life
 - Safety margins
- o Identify desirable materials
- o Identify 8,000 psi sensitivities
- o Resize components and distribution system for 8,000 psi
- o Estimate weight for baseline

To determine an overall system baseline weight, several advanced design techniques were incorporated. Acceptable performance levels as well as weight reduction were considered. The techniques used for this study were:

- o Force motors (all actuators)
- o Nonlinear valves
- o Asymmetric line loss
- o Local velocity reduction for water hammer suppression

Component weight changes with CTFE fluid at 8,000 psi were determined for the flight control, utility, engine nozzle actuators, heat exchangers and also the distribution system. Weights were derived using the 3,000 psi MIL-H-83282 production F-15 SMTD as a reference.

Figure 18 shows the 8,000 psi hydraulic system weight summary compared to the 3,000 psi system. The weight savings were 22 percent.

Baseline 8,000 psi F-15 SMTD Weight Summary		
Flight Control Actuators	325.73	
Engine Nozzle Actuators	284.36	
Utility Actuators	115.40	
Miscellaneous Components	436.25	
PC-1 and PC-2 Distribution System	92.66	MIL-H-83282
Utility Distribution System	105.26	Fluid
Fluid - CTFE	181.77	86.63
	<u>1,541.37</u>	<u>1,446.29</u>
	192.66	192.60*
	<u>1,734.03</u>	<u>1,638.89</u>
Reference 3,000 psi F-15 SMTD Weight Summary		
Flight Control Actuators	406.00	
Engine Nozzle Actuators	300.00	
Utility Actuators	147.30	
Miscellaneous Components	450.00	MIL-H-83282
Distribution Systems	347.80	Fluid
Fluid - CTFE	460.31	219.19
	<u>2,138.41</u>	<u>1,897.29</u>
	85.70*	85.70*
	<u>2,224.11</u>	<u>1,982.99</u>

*Note: Additional heat exchanger requirement.

Figure 18
3,000 psi and 8,000 psi Hydraulic System Weight Comparisons

The 8,000 psi actuators were sized using high strength materials and the latest recommended design factors. Distribution system sizing was achieved using the SSFAN computer program. Hydraulic lines were sized to achieve minimum weight and volume while maintaining an acceptable pressure loss. The CTFE fluid weight penalty was partially offset by the 8,000 psi system pressure which decreased actuator piston areas and distribution line sizes. The component weights and fluid volumes at 8,000 psi are shown in Figure 19.

	Dry Weight (lb)	Fluid Volume (in. ³)	Fluid Weight (lb) CTFE	Wet Weight (lb)
Jet Fuel Starter				
Valve				
JFS Manifold	6.52			6.52
Other	1.28			1.28
Miscellaneous				
JFS Hand Pump	3.00			3.00
Accumulator	35.98	111.88	7.16	43.14
Other	2.47	36.88	2.36	4.83
Subtotal	49.25	148.76	9.52	58.77
Aerial Refuel Receptacle				
Actuator	1.46	1.88	0.12	1.58
Valve	3.06			3.06
Subtotal	4.52	1.88	0.12	4.64
Air Induction				
Actuators				
Bypass Door	12.94	20.63	1.32	14.26
Diffuser Ramp	29.27	75.63	4.84	34.11
First Ramp	21.28	26.25	1.68	22.96
Other	2.04	1.09	0.07	2.11
Subtotal	65.53	123.60	7.91	73.44
Nozzle Controls				
Actuators				
Upper Rotating Vane	24.20	7.45	0.48	24.63
Lower Rotating Vane	24.20	7.45	0.48	24.68
Outboard Divergent Flap	84.52	56.04	3.59	88.11
Lower Divergent Flap	84.52	56.04	3.59	88.11
Convergent Flap	66.92	60.68	3.88	70.80
Valves	9.66	7.81	0.50	10.16
Subtotal	294.02	195.47	12.52	306.54
Canards				
Actuators	90.50	85.00	5.44	95.94
Valves	9.66	7.81	0.50	10.16
Subtotal	100.16	92.81	5.94	106.10
Ailerons				
Actuators	53.00	9.00	0.58	53.58
Valves				
Switching	9.66	7.81	0.50	10.16
Other	0.58			0.58
Subtotal	63.24	16.81	1.08	64.32
Stabilator				
Actuators	90.50	85.00	5.44	95.94
Valves				
Switching	9.66	7.81	0.50	10.16
Other	2.08			2.08
Subtotal	102.24	92.81	5.94	108.18
Arresting Hook Up/atch				
Actuator	2.29	5.63	0.36	2.65
Valve	0.99			0.99
Other	0.30			0.30
Subtotal	3.58	5.63	0.36	3.94
Main Landing Gear				
Actuator				
Retract	20.74	65.00	4.16	24.90
Uplock	6.64	6.72	0.43	7.07
Valves				
Uplock	0.45			0.45
Retract	12.51			12.51
Brake Operate	5.70			5.70
Emergency Ext	1.50			1.50
Miscellaneous	5.70	15.16	0.97	6.67
Subtotal	53.24	86.88	5.56	58.80
Ram Air Heat Exchanger Requirements				
	P.C. System	Utility System	Total	
Fuel/Oil H/X	9.1	9.1	18.2	
Ram Air H/X	13.4	13.7	27.1	
Fans	16.4	14.2	30.6	
Electric Motor	35.4	29.9	65.3	
Duct	9.0	7.5	16.5	
Installation	18.6	16.3	34.9	
Total	101.9	90.7	192.6	

	Dry Weight (lb)	Fluid Volume (in. ³)	Fluid Weight (lb) CTFE	Wet Weight (lb)
Nose Landing Gear				
Actuator				
Uplock	3.18	2.66	0.17	3.35
Retract	8.16	25.31	1.62	9.78
Steering and Damper	19.80			19.80
Valves				
	0.30			0.30
	6.70			6.70
	1.17			1.17
	1.17			1.17
Miscellaneous	0.20			0.20
Other	0.90	6.09	0.39	1.29
Subtotal	41.58	34.06	2.18	43.76
Emergency Generator				
Valve	4.32	4.06	0.26	4.58
Subtotal	4.32	4.06	0.26	4.58
Gun System				
Valve				
Gun Flow Regulator	2.34	6.72	0.43	2.77
Other	0.10			0.10
Subtotal	2.44	6.72	0.43	2.87
Rudder				
Actuator	38.73	15.63	1.00	39.73
Other	3.00			3.00
Subtotal	41.73	15.63	1.00	42.73
Flap/eron				
Actuator	53.00	9.00	0.58	53.58
Subtotal	53.00	9.00	0.58	53.58
ECS Auxiliary Air Inlet				
Actuator	2.94	1.88	0.12	3.06
Valve	1.26			1.26
Subtotal	4.20	1.88	0.12	4.32
Hydraulic Utility System				
Valve				
Temperature Regulator	2.04	6.25	0.40	2.44
Other	4.50			4.50
Miscellaneous				
Pump	64.00	80.00	5.12	69.12
Reservoir	22.90	348.00	22.27	45.17
Primary Heat Exchanger	0.83	0.63	0.04	0.87
Primary HX Valve	1.17			1.17
Other	41.73	43.44	2.78	44.51
Subtotal	137.17	478.32	30.61	167.78
Canopy				
Actuator				
Main	5.04	8.44	0.54	5.58
Lock	1.46	2.03	0.13	1.59
Other	0.35			0.35
Valve	3.33			3.33
Miscellaneous				
Accumulator	3.47	14.22	0.91	4.38
Other	0.35			0.35
Subtotal	14.00	24.69	1.58	15.58
Hydraulic PC-1 and PC-2				
Valve				
Temperature Regulator	4.08	12.50	0.80	4.68
Other	0.40			0.40
Miscellaneous				
Pump	64.00	80.00	5.12	69.12
Reservoir	16.60	224.00	14.34	32.94
Other	40.44	25.00	1.60	42.04
Subtotal	127.52	341.50	21.86	149.38
Total	1,161.74	1,680.51	107.57	1,269.31
Distribution System				
	Dry Weight (lb)	Fluid Volume (in. ³)	Fluid Weight (lb)	Wet Weight (lb)
PC1 and PC2	92.66	504.54	32.29	124.95
Utility	105.26	654.89	41.91	147.17
Total	197.92	1,159.43	74.20	272.12

Figure 19
8,000 psi Baseline
Hydraulic Equipment Weight

(1) Design Factors - For design of 8,000 psi hydraulic servocylinders, the following criteria were evaluated:

- o Burst pressure margin of safety
- o Proof pressure margin of safety
- o Fatigue life
- o Damage tolerance and rip-stop construction
- o Corrosion resistance
- o Cylinder breathing
- o Servocylinder stiffness
- o Qualification testing

Burst pressure factors have classically been assigned (for 3,000 psi service) as 2.5 times nominal operating system pressure and higher. The proof pressure factor for 3,000 psi systems is 1.5.

Figure 20 compares the Navy Lightweight Hydraulic System (LHS) Program (Reference 2) 8,000 psi design factors and the Flight Worthiness of Fire Resistant Hydraulic Systems (FWFRHS) program factors to 3,000 psi design factors. For this study, the design factors have been further modified as shown in Figure 21. The LECHT program criteria and the rationale for modification are presented in Figure 22. The reduction in transient pressure was introduced as a result of active and ongoing initiatives in the area of pump pulsations and water hammer control. The primary requirement is that the component shall be designed by infinite fatigue life when cycling between 8,800 psi peak transient pressure and the specified minimum pressure. Both the proof and burst pressure shall be consistent with the fatigue life requirement. The pressures and ratios shown in Figure 22 comply with that requirement. Nondestructive inspection (NDI) techniques have matured to the extent that material flaws can be detected early in the manufacturing process through ultrasonics, dye penetrant, magnaflux, and X-ray. This results in increased confidence in component structural integrity.

Component Pressure	3,000 psi System	Navy LHS 8,000 psi System	FWFRHS 8,000 psi System
Operating	3,000	8,000	8,000
Burst	7,500	16,000	18,000
Proof	4,500	12,000	11,000
Peak Transient	4,050	9,600	9,600
Tubing Auto Fretage Pressure	12,000	24,000	24,000

Figure 20
Design Criteria Comparison

	Proof	Burst	Transient	Impulse Cycling
Flight Control Actuator Pressure Side	10,000	16,800	8,800	1,600 to 8,800
Utility Component Pressure Side	10,000	16,800	8,800	1,600 to 8,800
Component Return Side	8,000	12,000	8,000	0 to 8,000
Tubing Pressure Side	Auto Frettag @ 24,000		8,800	0 to 8,800
Tubing Return Side	Auto Frettag @ 12,000		8,000	0 to 4,500

Figure 21
LECHT 8,000 psi Design Criteria

- Infinite Fatigue Life (10^7 Cycles)
- Fracture Tolerance
- Qualification With Damage
- Water Hammer Control
- State-of-the-Art NDI
- Corrosion Resistant Materials
- Burst Pressure/Operating Margin With
Variable Pressure Pump
 - Factor = 2.1 7% of Life
 - Factor = 5.6 93% of Life

Figure 22
Rationale for LECHT Design Criteria

Current technology hydraulic components are designed and qualification tested for adequate fatigue life under exposure to pressure transients from pump ripple and water hammer. Qualification requirements can be increased an order of magnitude to assure infinite fatigue life and minimum weight penalty.

Damage tolerant requirements have been imposed on many recent design activities. This calls for materials having superior fracture toughness and resistance to crack growth. A "rip stop" design using multipiece components is redundant when using fracture tough design. The criteria assumes finite flaws in critically stressed areas and analysis to assure that such a flaw will not propagate to fracture over the life of the component. The criteria is further modified, depending on the application, to allow detectable external leakage prior to reaching critical failure stress.

Corrosion from environmental conditions has been a problem with aluminum and carbon steels, but concern can be eliminated with the use of corrosion resistant steels and titanium. A materials review resulted in a candidate list of materials which have corrosion resistance, high strength, good fracture toughness and high fatigue allowable stress levels. These materials are presented in Figure 23. Titanium forgings are the optimum material for valve bodies and manifolds, as shown in Figure 24.

Material	F_{tu} (ksi)	F_{ly} (ksi)	F_a (ksi)	K_{IC} ($\sqrt{\text{in./in.}}$)	ρ (lb/in. ³)	Comments
Ti 10-2-3 STA	180	169	125	49	0.168	High Strength - Less Fracture Toughness
Ti 10-2-3 BAOA	140	130	77	100	0.168	6Al-4V-ELI Is 115/105/ $K_{IC} = 80$
PH 13-8Mo H1000	205	190	150	90	0.279	Currently in Use
15-5 PH H1025	155	145	130	100+	0.283	Popular Material in Extensive Use
Carpenter Custom 455	220	205	115	73	0.280	Values at Cond H950 $K_{IC} = 100$ at Cond H1000
AF1410	235	220	150	130+	0.280	Not Corrosion Resistant; in Limited Use
Al-Li High Strength	(70 - 76)	(58 - 68)	—	31	0.093	Development Alloy Moving Strong

Figure 23
Candidate Component Materials

- Structurally 36% Lighter Than Aluminum
- Use Fracture Tolerant Design Criteria in Lieu of "Rip Stop" Construction
- Optimum Forging Alloy Ti 10-2-3
- Higher Fabrication Cost/Higher Reliability

Figure 24
Titanium Components

Cylinder breathing criteria has been established for 3,000 psi cylinders with radial expansion limits to prevent piston seal leakage at pressure. These limits are imposed at current burst pressure limits with existing piston seal and gland designs. Cylinders designed for 8,000 psi having a burst pressure factor of 2.1, are marginal for the current breathing criteria. Since new seal technology is emerging for 8,000 psi and CTFE fluid, new criteria may be needed for cylinder wall thickness, seals and glands.

Using the 3,000 psi proof pressure factor of 1.5 at 8,000 psi, makes cylinder design as a pressure vessel excessively heavy. A lower proof pressure still ensures seal integrity, tightness of fits and fastener preload.

Servocylinder axial stiffness (spring rate) is reduced with the lower CTFE fluid bulk modulus and smaller cylinder piston diameters at 8,000 psi. The magnitude of the reduction is such that changes in operating geometry (hinge arm lengths) and enhanced dynamic stiffness may be required.

Relaxation of the burst pressure margin should enhance qualification testing. Life cycling would be increased to an order of 10^7 to demonstrate infinite life. It also is reasonable to test for damage tolerance as well. This can be accomplished by performing some portion of the "pressure vessel" life cycling with known flaws such as dings, scratches, or improperly torqued bolts. This approach makes it practical to define allowable damage in service to avoid scrapping parts which are still serviceable.

(2) Component Sizing - Figure 25 presents a list of hydraulic servo-actuators resized for 8,000 psi. Resizing was accomplished by maintaining existing rod design, retracted length and stroke, as shown in Figure 26. Optimization for flight application would require reevaluation of loads and hinge moments.

The design factors used to resize equipment were limited to those which could be applied universally. Design criteria which were used for weight comparisons included:

- o Axial stiffness
- o Elastic flexural analysis
- o Cylinder breathing
- o Fracture analysis
- o "Rip stop" construction
- o Vibration and 'g' loads

With 8,000 psi, the reduced flow rates reduced the size and weight of the distribution system. Actuator imbalance (extend/retract volume delta) was minimized to reduce reservoir volume. Minimum imbalance was achieved in balance tube type actuators using the minimum rod wall thickness designed for external burst pressure stability, considering column bending and stepped beam column stability. When it was necessary to increase rod wall thickness above the minimum because of other criteria, imbalance and extension flow were affected accordingly. Figure 27 shows the effect on actuator flow rates due to increased rod wall thickness on a set of convergent nozzle actuators.

Actuator	No. Per A/C	Bore	Stroke	Rod Dia	Center Rod Dia	Balance Rod Dia	Area (in. ²)		Output Force		Flow Rate (GPM)		Δ Vol (in. ³)
							Ext	Ret	Ext	Ret	Ext	Ret	
Stabilator/Canard	4	2.5738	7.771	1.684	1.9728	1.684	2.9755/ 2.1461	2.1461/ 2.9755	40,460	40,460	6.5/ 4.68	4.68/ 6.5	6.45/ 6.45
Aileron/Flaperon	4	2.3954	1.37	1.85	2.1606	1.85	1.8186/ 0.8402	0.8402/ 1.8186	21,000	21,000	1.58/ 0.73	0.73/ 1.58	1.84/ 1.84
Rudder	2	—	± 30°	—	—	—	Rotary Actuator		± 22,000 in.-lb		2.23	2.23	—
First Ramp	2	1.7176	8.85	1.209	—	—	2.317	1.169	18,420	9,120	19.02	6.98	10.16
Diffuser Ramp	2	2.1566	10.17	1.48	—	—	3.653	1.933	29,000	15,100	14.19	5.61	17.5
Bypass Door	2	2.3853	2.67	0.934	—	—	1.873	1.188	14,870	9,320	8.38	4.38	1.83
Converging Nozzle	4	1.797	10.2	1.25	—	1.105	1.579	1.309	12,500	10,300	3.67	3.05	2.75
Diverging Nozzle	8	1.916	15.2	1.62	—	1.430	1.276	0.822	10,100	6,450	3.25	2.09	6.89
Targeting Vanes	4	1.677	3.80	1.25	—	1.105	1.250	0.9804	9,900	7,720	0.32	0.26	1.03

Notes:

- 1) All dimensions are in inches unless otherwise noted
- 2) Flow rates quoted are maximum no-load flows without electronic limiting
- 3) Dual tandem cylinders are balanced by equal opposing imbalance

Figure 25
Hydraulic Actuator Data

- Carry Over From Baseline
 - Output Forces and Rates
 - Retracted Length and Stroke
 - Output Rod and Bearing Sizes
- Resize for 8,000 psi
- Estimate Required Wall Thickness
 - Infinite Fatigue Life
 - 12,000 psi Minimum Burst Pressure
 - High Strength Materials
- Estimate New Flow Rates and Weight

Figure 26
Component Resizing Techniques

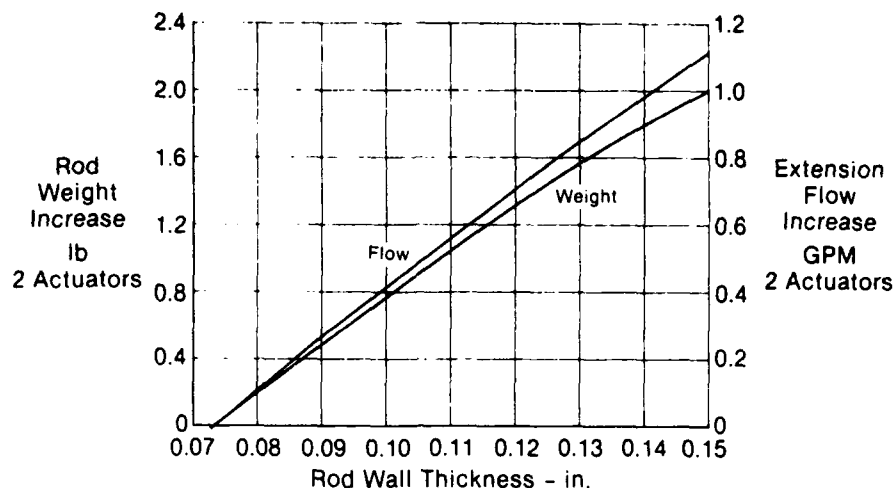


Figure 27
Sensitivity of Imbalance to Rod Thickness

Actuator cylinder bore diameters were determined using 7,900 psi across the piston holding the stall load for certain applications. Where a load at rate is the design criteria (e.g., engine inlet ramp actuators and engine nozzle actuators), analysis was performed to determine the optimum pressure differential to be supplied at load flow. Figure 28 shows that with

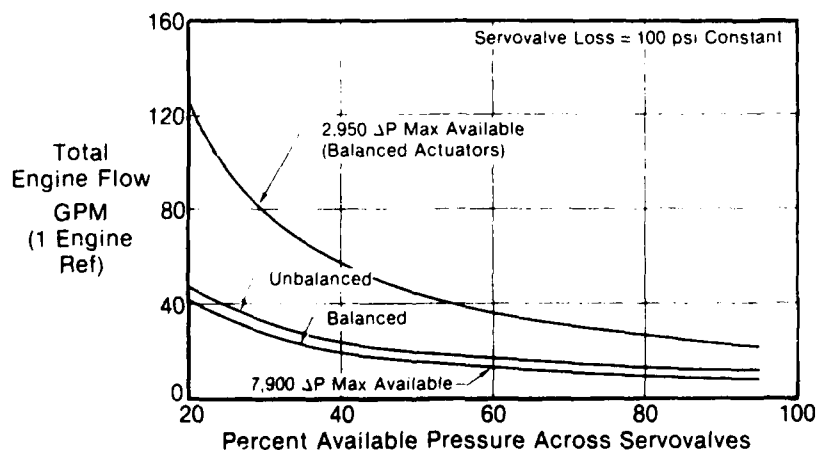
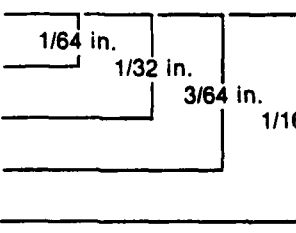


Figure 28
Maximum Engine Hydraulic Load

increasing pressure differential, the required flow rate decreased to an asymptotic value. In the case of the nozzle actuators, 80 percent of the available system pressure was selected for the baseline (low flow, small distribution line, minimum reservoir volume). This pressure differential was exercised in evaluating component weights.

Minimum wall thickness criteria selected for cylinder barrels was the greater of that required for infinite fatigue life, or a burst pressure of 12,000 psig. In cases where servovalve manifolding could be integrated with the servocylinder tailstock, stainless steels or titanium were used.

Design sensitivities which present a significant effect at 8,000 psi, included standard O-ring sizing. The current practice of sizing a cylinder bore and rod and then rounding up to the next larger O-ring size had an unacceptable effect on actuator force output and flow rates. This was not applied in the 8,000 psi sizing. For future design, it will be necessary either to use nonstandard seals and glands, or create standards intermediate to existing sizes. This could be implemented by expanding the standards size band with a unified approach, as shown in Figure 29.

MS 28775 Dash No.	Piston Dia	Oversize Ref	Force Output* (lb)
-125	1.489		13,892
-125A	1.5046		14,185
-125B	1.520		14,478
-125C	1.5359		14,783
-126	1.551		15,076

*Output force with 1.00 in. rod diameter at 8,000 psi

Figure 29
Intermediate O-rings

Figure 29 also illustrates the force output variation at 8,000 psi for the O-ring size applied to an unbalanced cylinder with a one inch rod diameter and the cylinder bore variable. In this instance, with the use of intermediate sizes, the force output compromise would be reduced from over 1,000 lb to less than 300 lb.

Weights of servocylinders were calculated using (for baseline) linear or rotary force motors driving shrink fitted servovalves. Shrink fit of servovalves eliminated many O-ring seals and reduced the valve diameter and length. Force motor weights used were conservative and can be expected to improve with future development of flight weight force motors. Sixteen force motors were used on the engine nozzle actuators and eight on the flight control actuators.

Figure 30 shows the resized 8,000 psi stabilator/canard actuator superimposed on the actuator 3,000 psi envelope. Size and weight were both decreased. Actuators having significantly larger rod sizes, such as aileron/flaperon actuators where the size is driven by vibration 'g' loads in the wing, have much less reduction. In the stabilator/canard cylinder, the

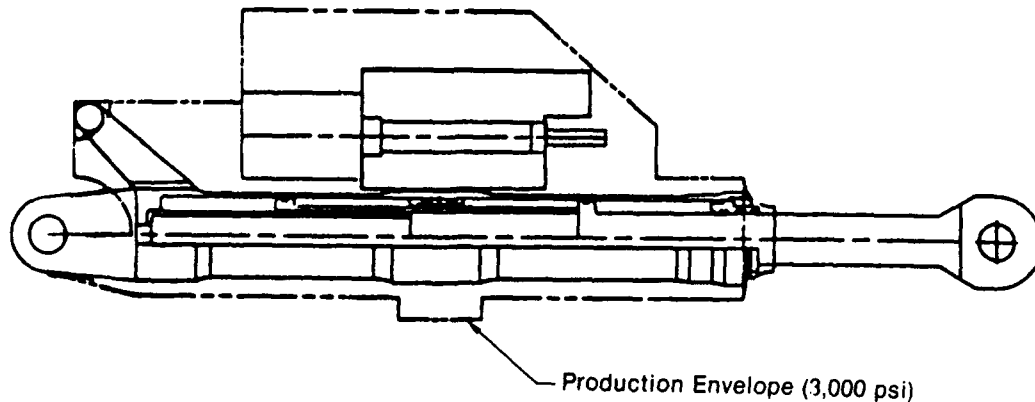


Figure 30
Stabilator/Canard Actuator

bore was reduced 23.6 percent while the aileron/flaperon bore was reduced only 13.8 percent. In applications such as these, weight savings are less, and are achieved primarily from high strength materials and lower burst pressure margins.

Distribution system components, such as the reservoirs and filter manifolds, do not resize in the same manner as actuators. The manifold weight and volume is driven by the degree of filtration, flow rate, required time between servicing, system pressure (both pressure and return), and construction materials. The reservoirs in the F-15 SMTD aircraft are Reservoir Level Sensing (RLS) and have a high pressure valving head which can accommodate the total pump output flow and pressure. For these reasons, the manifold and reservoir were evaluated for other system effects after the basic design was completed.

(3) Distribution System Sizing - The main tool used in sizing the distribution systems was the SSFAN computer program.

Models for the LECHT hydraulic system were derived from previously developed 8,000 psi CTFE F-15 models for the Air Force contract Flight Worthiness of Fire Resistant Hydraulic Systems (FWFRHS). Actuators were resized and canard and nozzle systems were added to make these models representative of an 8,000 psi F-15 SMTD. The trailing edge flap actuators were changed to variable position flaperon actuators.

Several ground rules of the LECHT program affected the tube sizing. The asymmetric pressure drops necessitated smaller tubes on the pressure side and larger tubes on the return. Tube sizes were increased at control valve inlets to limit inlet velocities to 30 feet per second which reduced waterhammer transients. Odd tube sizes were used on the pressure side and even sizes on the return. An exception was made for thick-walled 3/16 inch diameter tubing, which was used on both pressure and return. Figure 31 summarizes the tubing data.

O.D. (in.)	Wall (in.)	I.D. (in.)	Flow Area (in. ²)	Wet Weight (lb/ft)	Minimum L Between Bends	Minimum Bend Radius	Usage
3/16	0.020	0.1475	0.0171	0.0340	1.00	0.75	Pressure and Return
3/16	0.039	0.1095	0.0094	0.0430	1.00	0.75	Coil (Pressure and Return)
1/4	0.016	0.218	0.0373	0.0523	1.00	0.75	Return
1/4	0.026	0.1980	0.0308	0.0600	1.00	0.75	Pressure ⁽³⁾
1/4	0.028	0.1940	0.0296	0.0614	1.00	0.75	Return Coil or Pressure
1/4	0.053	0.144	0.0163	0.0770	1.00	0.75	Pressure Coil
5/16	0.032	0.2485	0.0485	0.0933	1.00	1.25	Pressure
5/16	0.066	0.1805	0.0256	0.1201	1.00	1.25	Pressure Coil
3/8	0.019	0.337	0.0891	0.1117	1.12	1.125	Return
3/8	0.042	0.2910	0.0665	0.1382	1.31	1.125	Return Coil or Pressure
7/16	0.045	0.3475	0.0948	0.1831	1.31	1.32	Pressure
7/16	0.092	0.2535	0.0505	0.2350	1.31	1.32	Pressure Coil
1/2	0.026	0.448	0.1576	0.1996	2.00	1.50	Return
1/2	0.056	0.388	0.1182	0.2457	2.00	1.50	Return Coil or Pressure
9/16	0.058	0.4465	0.1566	0.3029	1.69	1.69	Pressure
9/16	0.119	0.3245	0.0827	0.3893	1.69	1.69	Pressure Coil
5/8	0.032	0.561	0.2471	0.3109	2.50	1.875	Return
5/8	0.071	0.4830	0.1832	0.3857	2.50	1.875	Return Coil or Pressure
11/16	0.071	0.5455	0.2337	0.6036	2.50	2.13	Pressure
3/4	0.039	0.672	0.3546	0.4492	2.50	2.25	Return
13/16	0.084	0.6445	0.3262	0.6325	2.50	2.50	Pressure
1	0.051	0.898	0.6333	0.7952	3.00	3.00	Return

Notes

- (1) Wall thickness of new tube sizes for pressure service are based on an autofrettage pressure of 24,000 psig for 3Al-2.5V CWSR 105
(2) Wall thickness of new tube sizes for pressure coil service are based on 3Al-2.5V CWSR 105 working at 48,000 psi maximum
(3) Theoretical minimum wall size for 8,000 psi, unlikely to be developed in actual practice

Figure 31
Titanium Tubing Data (3Al-2.5V)
CTFE at 8,000 psi

The systems were sized with flight control actuators operating at maximum rate and no load. Other systems, including engine nozzle and inlet, gun, radar, etc., were sized at their load flow rates. The PC systems were sized to provide flow to the nozzle controls with no degradation of performance.

The largest lines in the PC systems are 3/4 inch diameter suction lines just upstream of the pump. The largest pressure lines are 11/16 inch diameter tubes at the pump outlet. There are 3/16 inch diameter tubes used in return lines from the aileron and flap/aileron actuators. Tube sizes and lengths are shown in Figure 32.

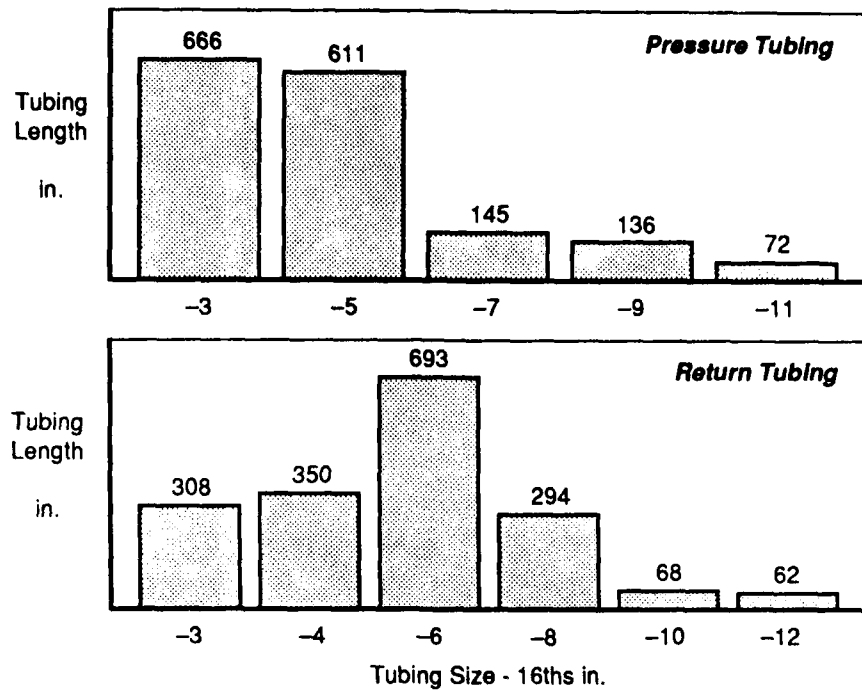


Figure 32
PC System Pressure and Return Tubing

Figure 33 shows line sizes of the connections between components for a PC system. Unlabeled segments have negligible length. The largest lines are around the pump and at the inlets of the stabilators and canards. Low flow in the ailerons and flaperons enabled 3/16 inch diameter lines to be used on both the pressure and return sides.

The utility system lines are larger because of the higher flow rates and the greater pressure drop required across the engine nozzle and inlet actuators. The largest utility system lines are 13/16 inch diameter on the pressure side and 1 inch diameter on the return. Line lengths and sizes for the utility system are shown in Figure 34.

Figures 35, 36, and 37 show the line sizes between components for the utility system. The large trunk lines in Circuit A allowed the use of 3/16 inch diameter tubes in much of the rest of the circuit. Unlike the PC system model, large tubes are not required at the control valve inlets because maximum flow rates are lower for the utility system actuators.

The computer program used to weigh the distribution systems included factors for end fittings, clamps, unions and coil tube blocks, as well as the weights of the actual tubes. The factors are based on actual weights of the production F-15 distribution system. The distribution system weight results are shown in Figure 38.

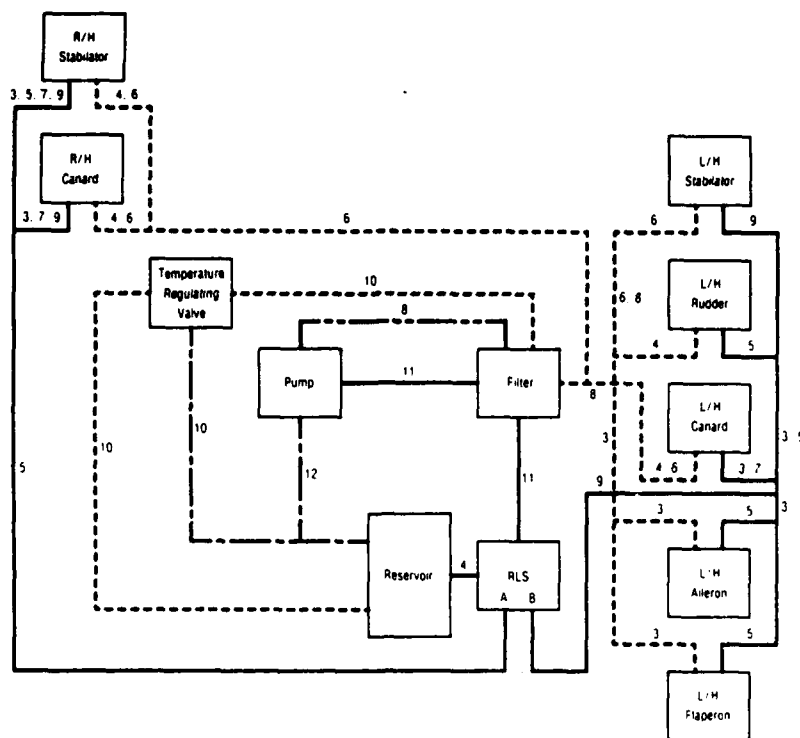


Figure 33
PC-1 Hydraulic System

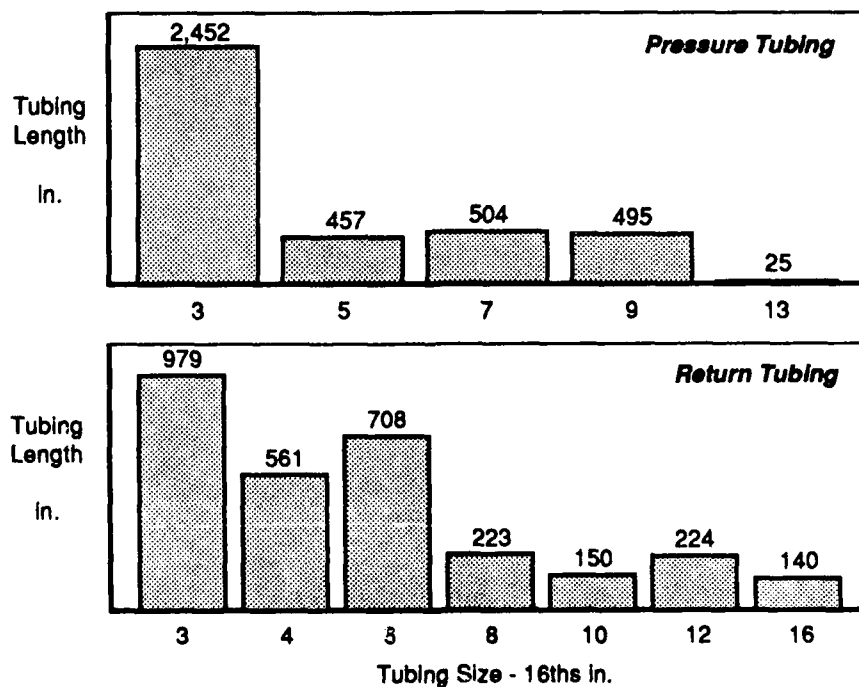


Figure 34
Utility System Pressure and Return Tubing

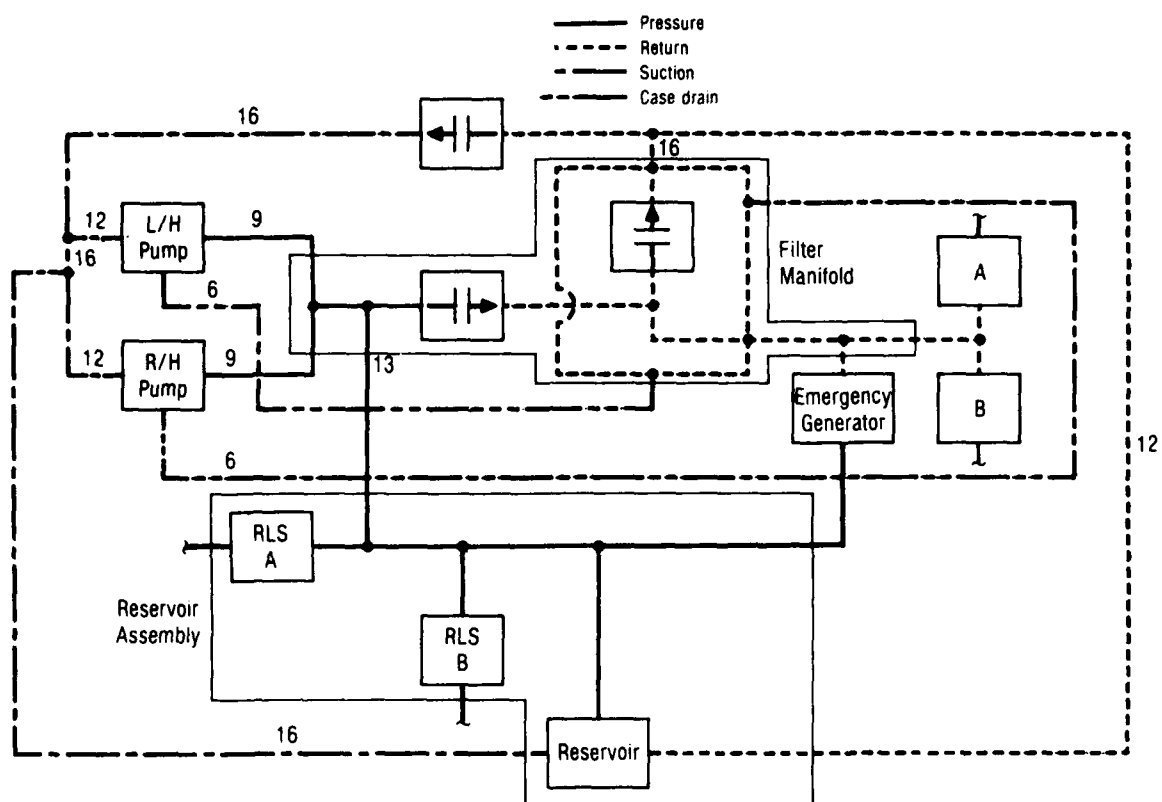


Figure 35
Utility Central Hydraulic System

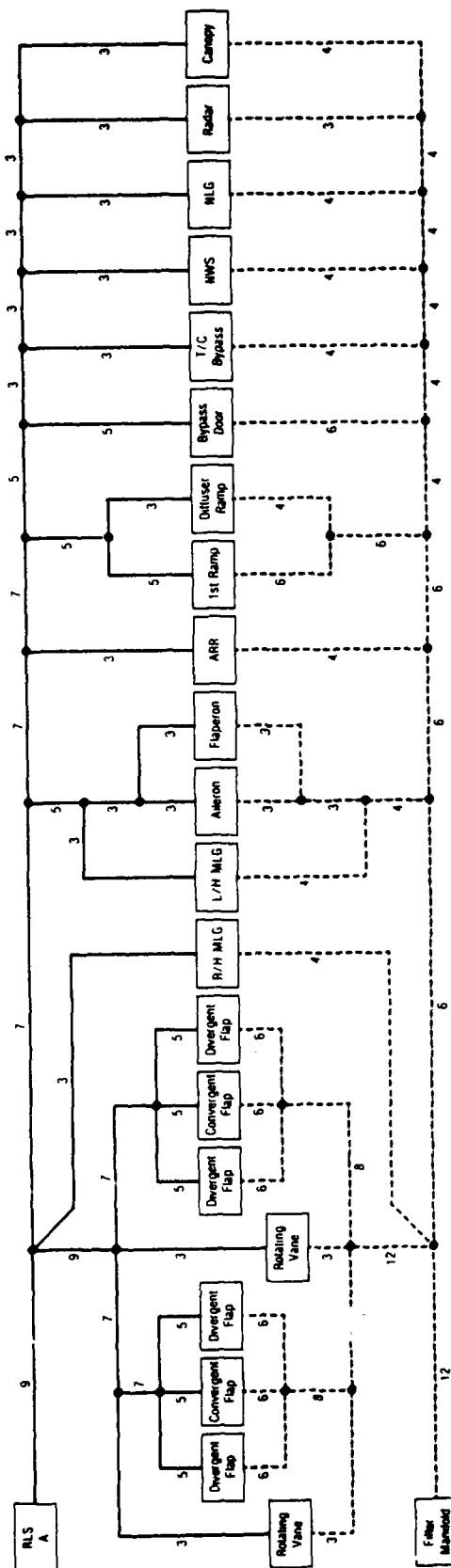


Figure 36
Line Sizes
 LECHT SMTD 8,000 psi
 Baseline Utility System Circuit A

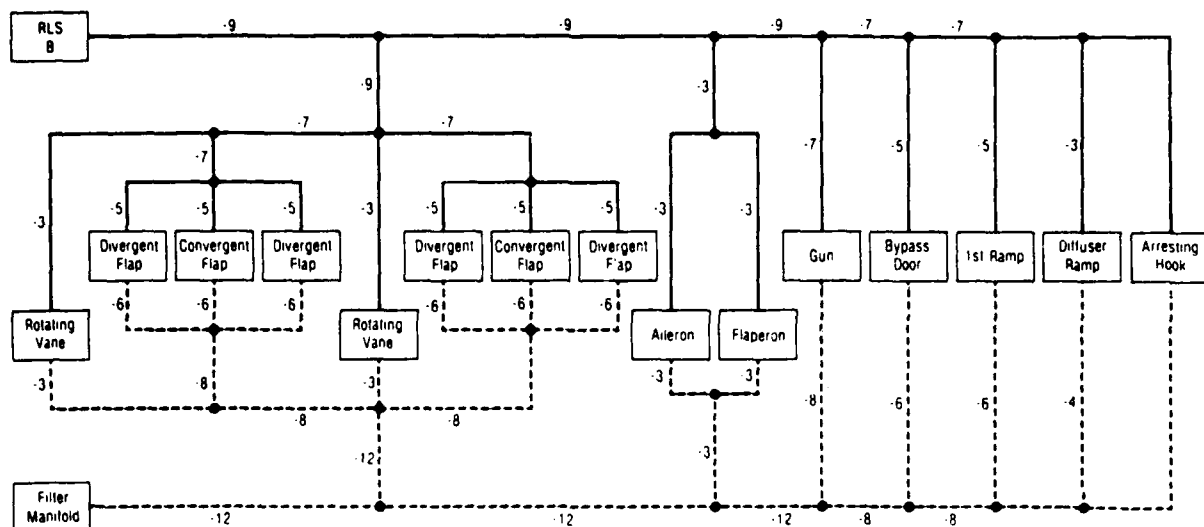


Figure 37
Line Sizes
 LECHT SMTD 8,000 psi
 Baseline Utility System Circuit B

		One PC System	Utility System	Utility System + 2 PC Systems
Dry Weight	(lb)	45.60	103.29	194.49
Fluid Volume	(in. ³)	252.27	654.89	1,159.43

Figure 38
 Distribution System Weight/Volume Summary

b. Hydraulic System Thermal Analysis - A thermal analysis was performed on the baseline 8,000 psi F-15 SMTD hydraulic systems. The ground rules and thermal design criteria are listed below:

- o Fuel Temperature to Engine less than 195°F
- o Fuel Tank Temperature less than 135°F
- o Hydraulic Fluid Temperature less than 275°F
- o No Modifications to Aircraft Fuel System (Use Existing Fuel Recirculation Capability)
- o Add Ram Air Heat Exchangers as Necessary
- o No Air Circulation in Engine Compartment
- o No Attempt Made to Control Hydraulic Fluid Temperature to Engine Nozzle Actuators
- o Eight Engine Nozzle Actuators Lumped into One Model for Flow Rates and Surface Areas

The first task was to perform a hydraulic thermal analysis on a production version of the 3,000 psi SMTD aircraft to establish a reference. A similar hydraulic thermal analysis was then performed on the baseline F-15 8,000 psi SMTD aircraft. Temperature level deltas were established, and the heat rejection and additional heat exchanger weight at 8,000 psi were determined. After determining 8,000 psi heat rejection, additional fuel/oil and ram air type heat exchangers were sized for use in the LCC study.

(1) Computer Model/Approach - The MCAIR thermodynamics Heat Transfer Program was used to determine fluid temperature levels in the hydraulic system during a typical Short Take-Off and Landing (STOL) Mission Flight. Figure 39 illustrates the hydraulic system/fuel system heat exchanger

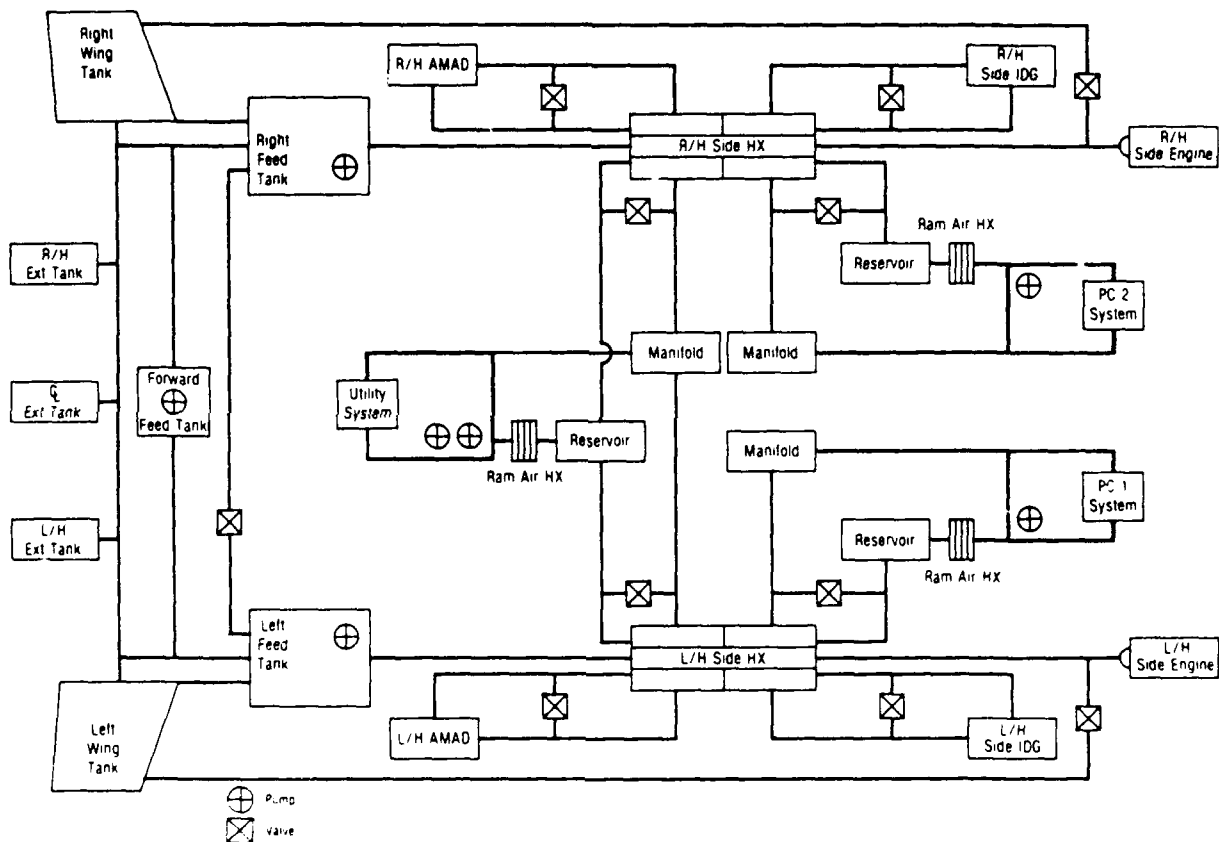


Figure 39
F-15 SMTD Fuel/Hydraulics System Interface

interface. The transient thermal models developed for the Utility, PC-1, and PC-2 hydraulic systems, are shown in Figures 40, 41 and 42. These thermal models incorporate appropriate tubing diameters and lengths, materials, hydraulic flow rates and component heat rejection rates based on laboratory testing. The severe environment for the engine nozzle actuators was defined by the engine manufacturer, shown in Figure 43. Convection and radiation heat transfer to the ambient environment and to other components and the hydraulic fluid is considered in the model, as well as conduction through insulation on the engine nozzle actuators.

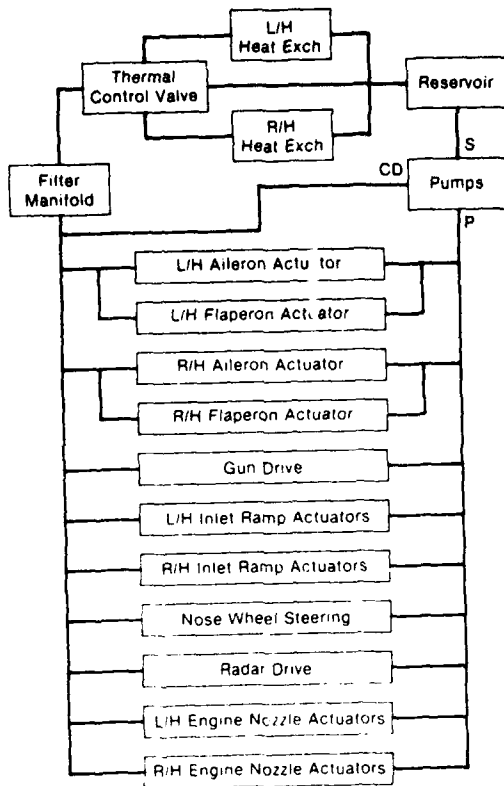


Figure 40
CTFE 8,000 psi F-15 SMTD Utility
Hydraulic System Thermal Model

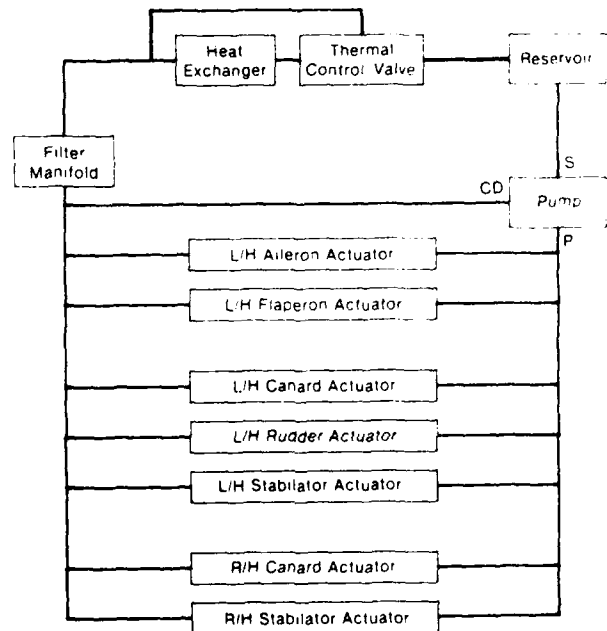


Figure 41
CTFE 8,000 psi F-15 SMTD
PC-1 Hydraulic System Thermal Model

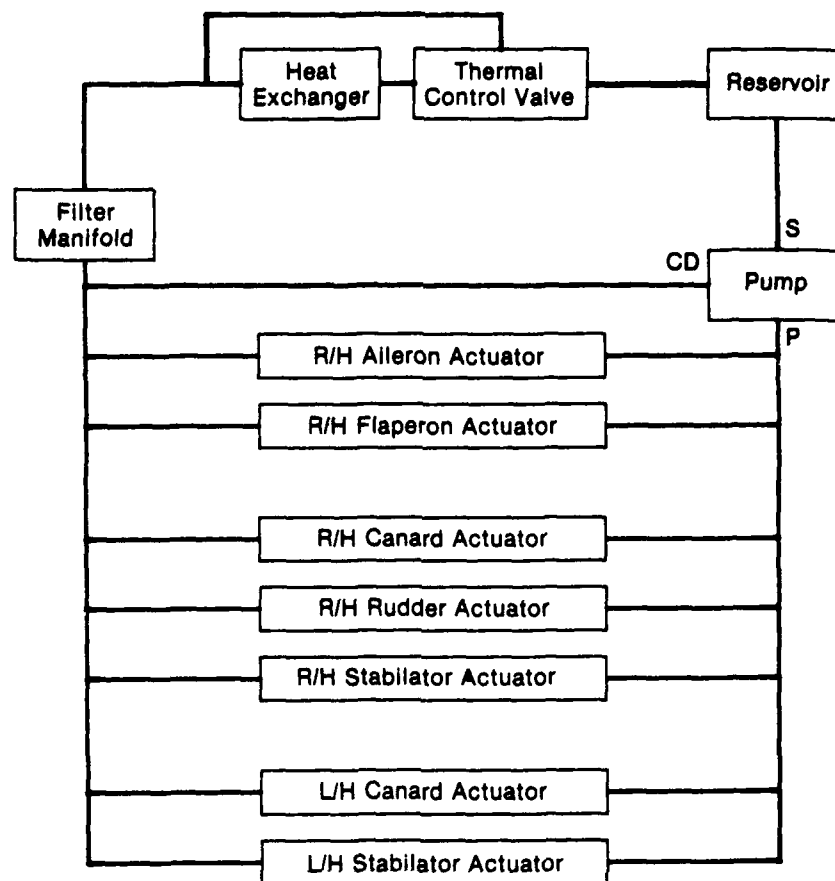


Figure 42
CTFE 8,000 psi F-15 SMTD
PC-2 Hydraulic System Thermal Model

- Engine Wall
 - 700°F Wall
 - 1/2 in. Insulation
 - Contact With 40% Actuator Surface Area
- Engine Compartment
 - 495°F Ambient Air
 - Natural Convection
 - 60% of Actuator Surface Area Exposed

Figure 43
Thermal Design Parameters
Engine Nozzle Actuators

Temperature histories were calculated for the hydraulic fluid at various points in the system, indicating the hydraulic temperature transients during the course of a typical STOL mission profile, see Figure 44. These temperature levels were used to determine the additional heat exchanger requirements to maintain the hydraulic fluid and fuel temperatures at an acceptable level. It should be noted that no attempt was made to control engine nozzle actuator temperatures.

Flight Phase	Time in Phase (sec)	Cumulative Time (sec)
Taxi	900	900
STOL	14	914
Climb and Retract Gear	399	1,313
Cruise	1,336	2,649
Combat	612	3,261
Cruise	2,557	5,818
Descent	1,072	6,890
STOL	180	7,070
Reverse	7	7,077

Figure 44
F-15 8,000 psi SMTD Mission Profile

The temperature vs. time histories for the 3,000 psi SMTD baseline with and without ram air heat exchangers are shown in Figures 45 through 56. Figures 57 through 68 show the temperature profiles with and without ram air heat exchangers for the 8,000 psi SMTD baseline.

(2) Baseline Hydraulic System Heat Rejection - Heat generation levels were determined using the heat rejection created by the system pump and the flow rates occurring at the actuators. The pump heat rejection was derived from published pump supplier data, see Figure 69. Heat added by system flows was derived from the sum of the average operational and leakage flow rates for each mission segment. Operational flow was established using percentages derived from F-18 flight test data. These percentages were used to adjust the F-15 SMTD flow rates.

The initial thermal analysis was used to determine the 3,000 and 8,000 psi F-15 SMTD heat generation using MIL-H-83282 and CTFE hydraulic fluids. The total heat generation was determined from several factors.

Valve null leakage was determined from laboratory testing and component supplier data. Initially, values chosen for the 3,000 and 8,000 psi null leakages did not incorporate digital noise effects. Later, it was decided to incorporate digital noise effects with the 8,000 psi hydraulic system.

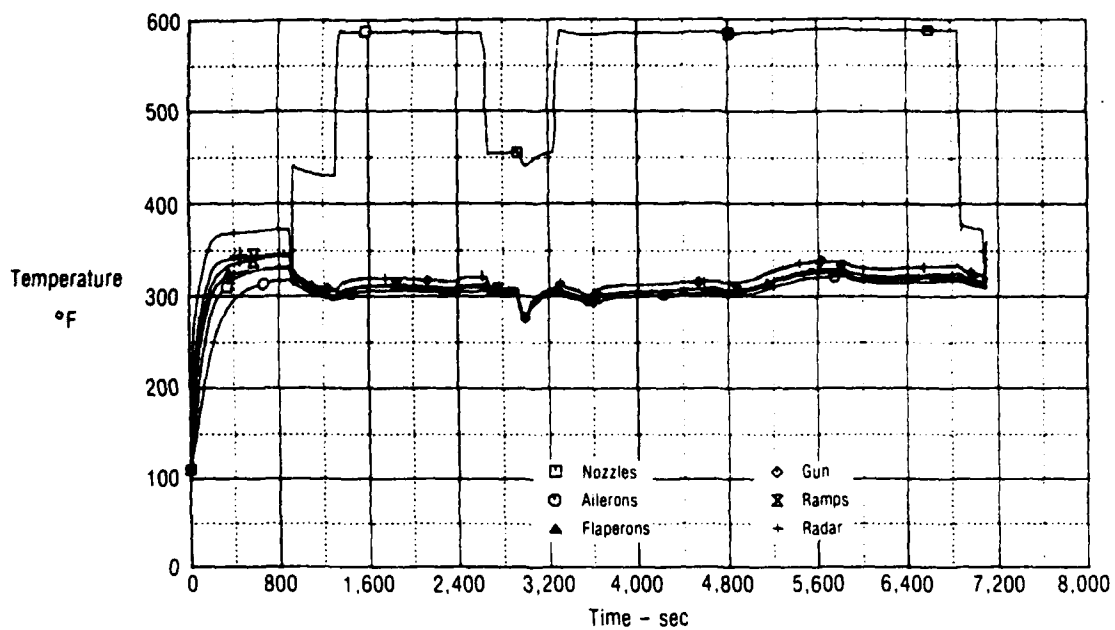


Figure 45
3,000 psi Baseline Without Ram Air HX
 Utility Actuator Exit Temperatures

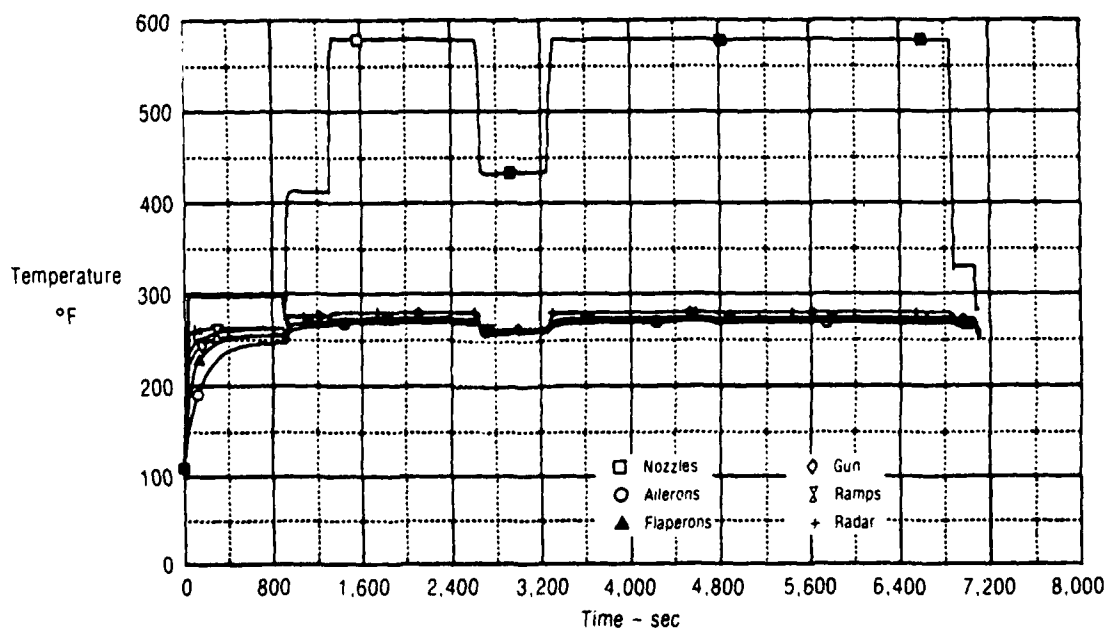


Figure 46
3,000 psi Baseline
 Utility Actuator Exit Temperatures

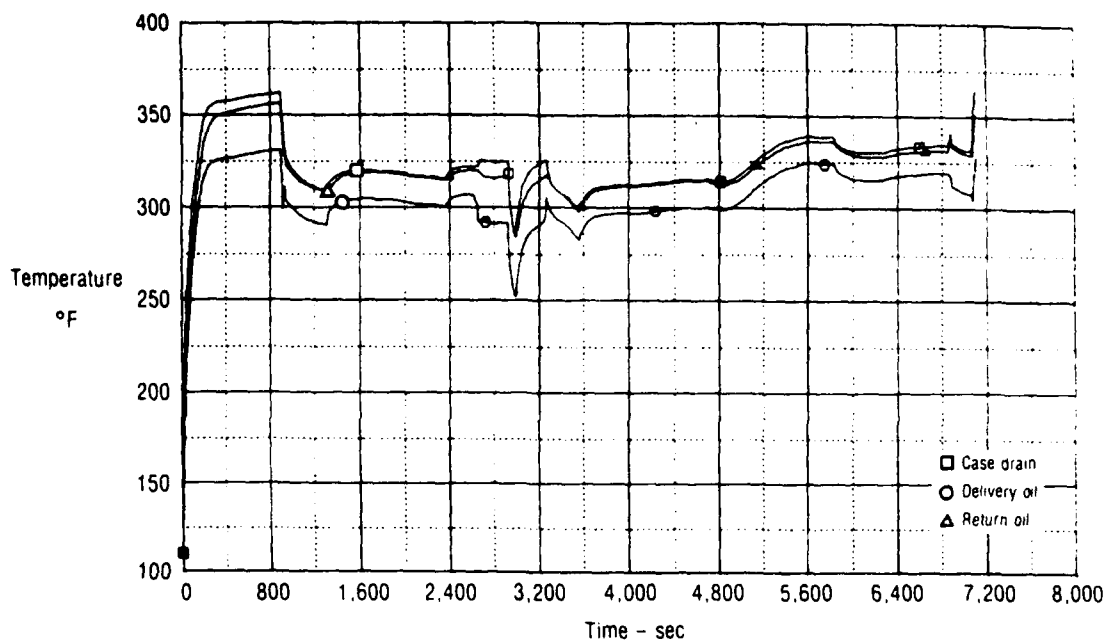


Figure 47
3,000 psi Baseline Without Ram Air HX
Utility Pump and Return Temperatures

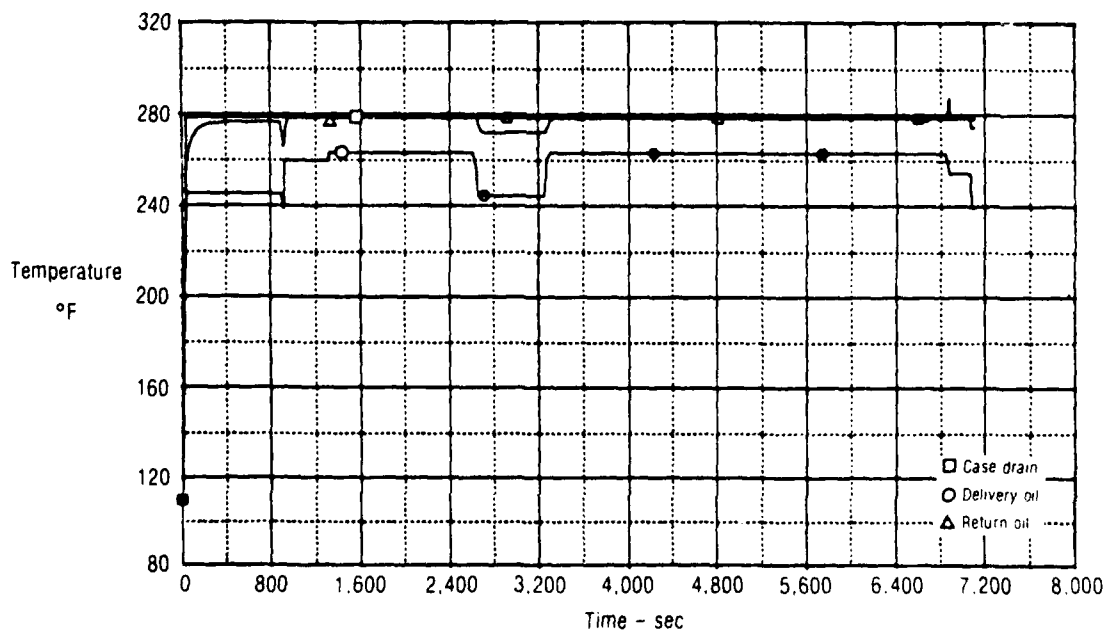


Figure 48
3,000 psi Baseline
Utility Pump and Return Temperatures

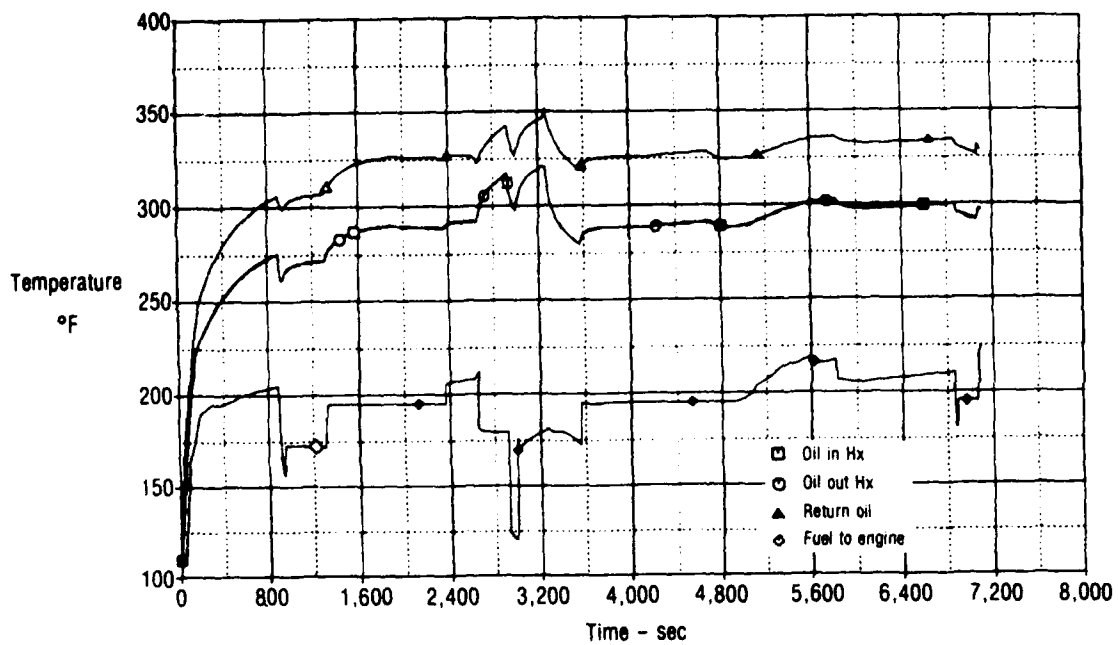


Figure 49
3,000 psi Baseline Without Ram Air HX
 Utility Ram Air HX Requirements

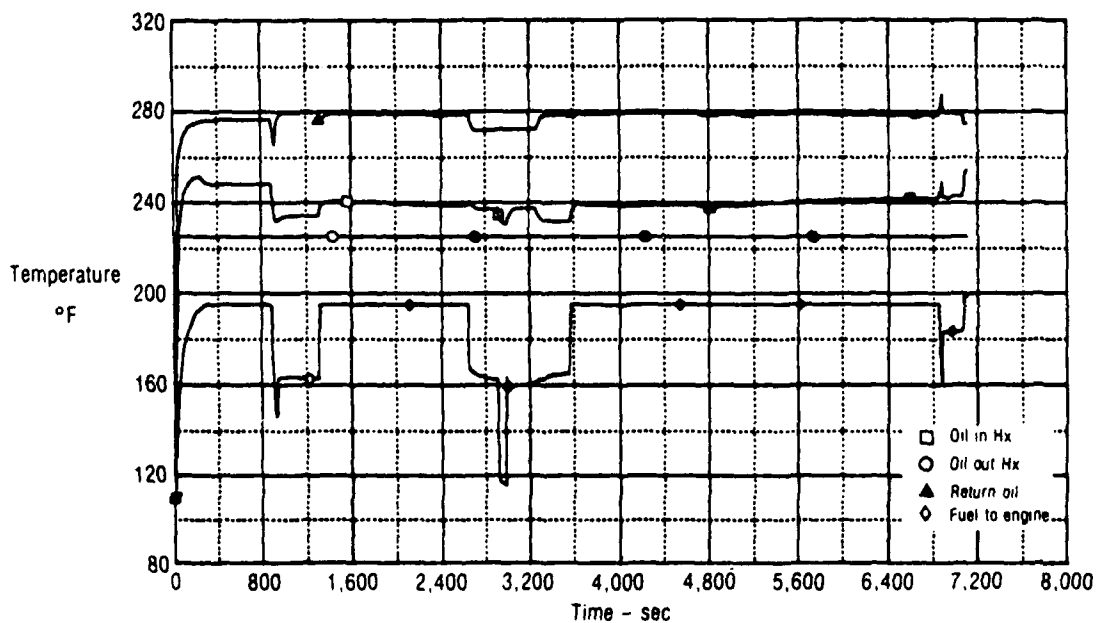


Figure 50
3,000 psi Baseline
 Utility Ram Air HX Requirements

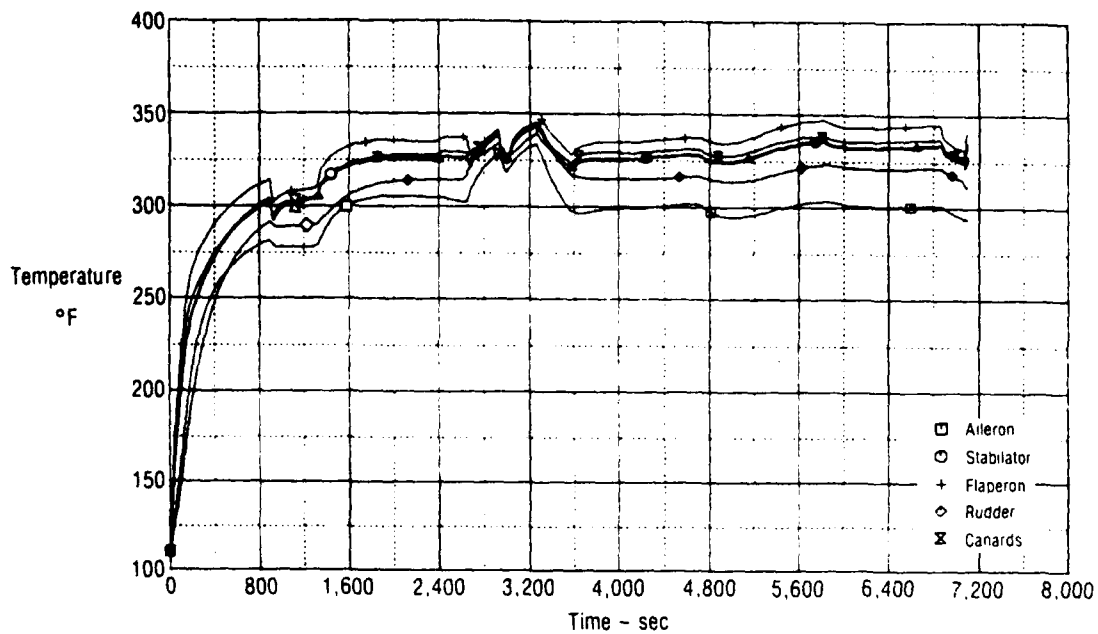


Figure 51
3,000 psi Baseline Without Ram Air HX
PC-1 Actuator Exit Temperatures

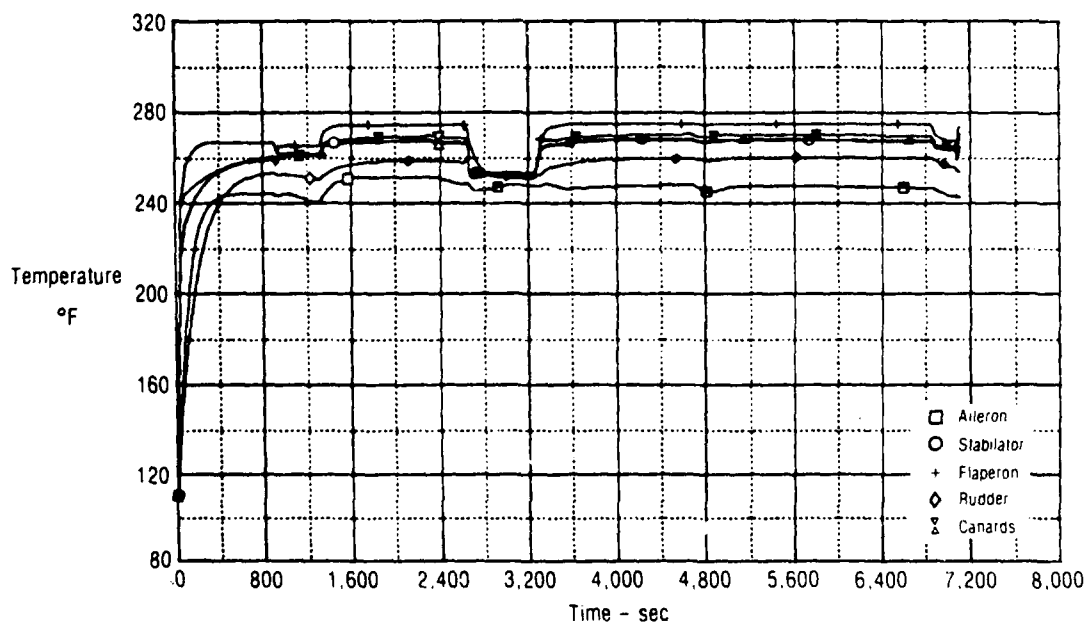


Figure 52
3,000 psi Baseline
PC-1 Actuator Exit Temperatures

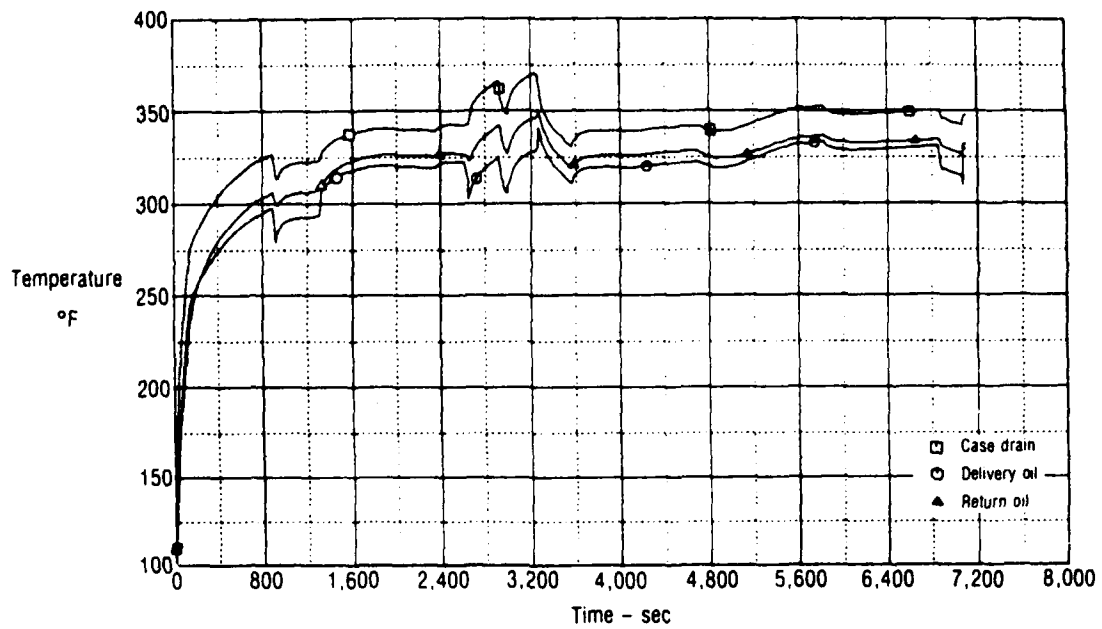


Figure 53
3,000 psi Baseline Without Ram Air HX
PC-1 Pump and Return Temperatures

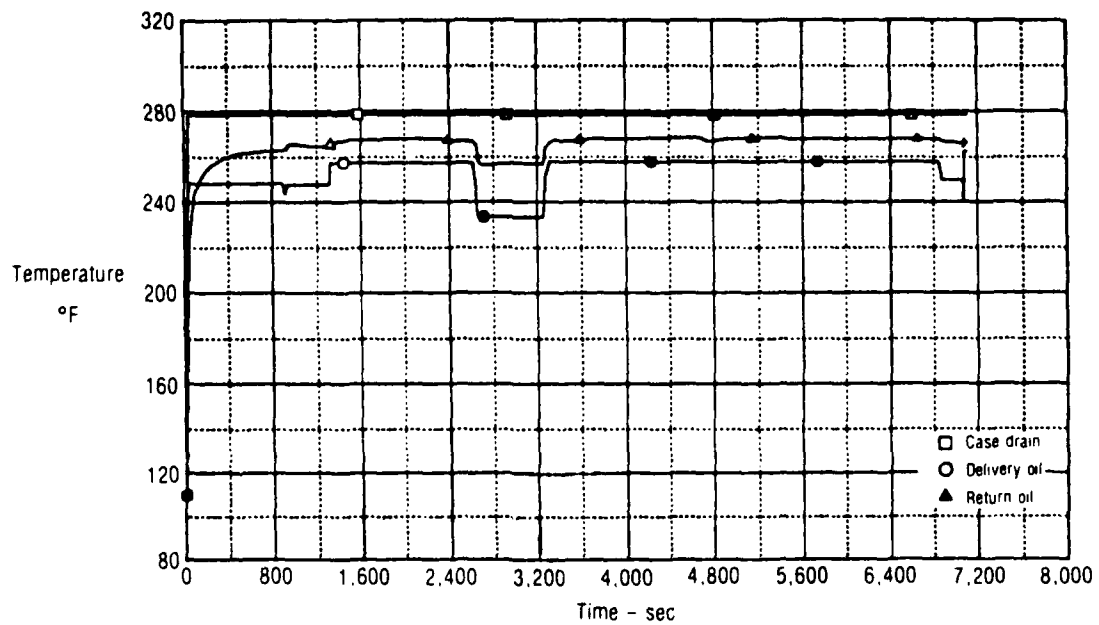


Figure 54
3,000 psi Baseline
PC-1 Pump and Return Temperatures

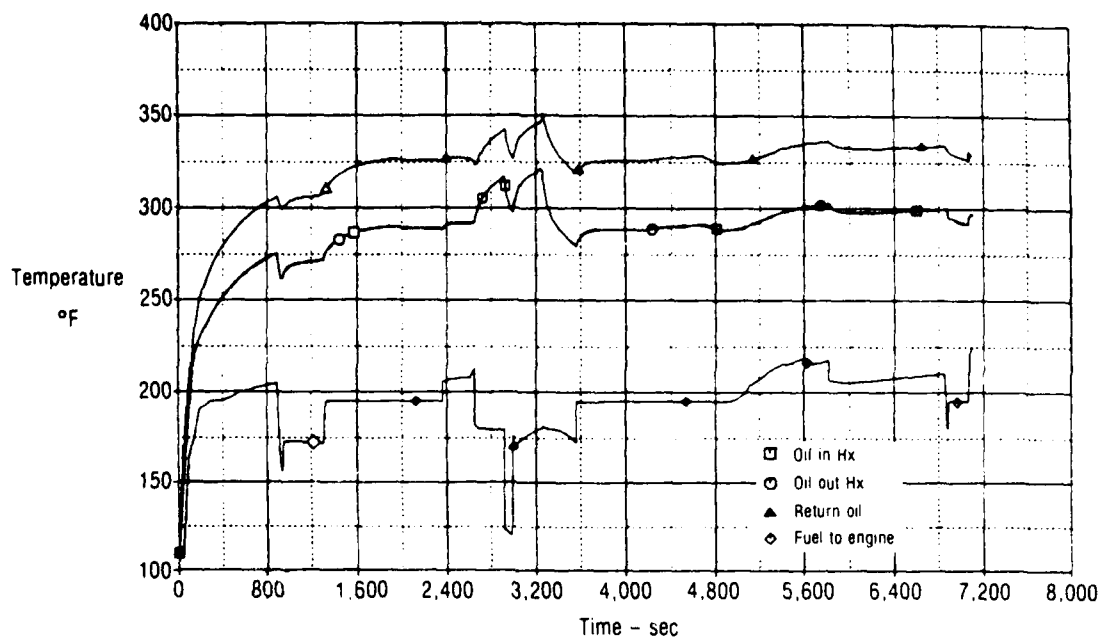


Figure 55
3,000 psi Baseline Without Ram Air HX
PC-1 Ram Air HX Requirements

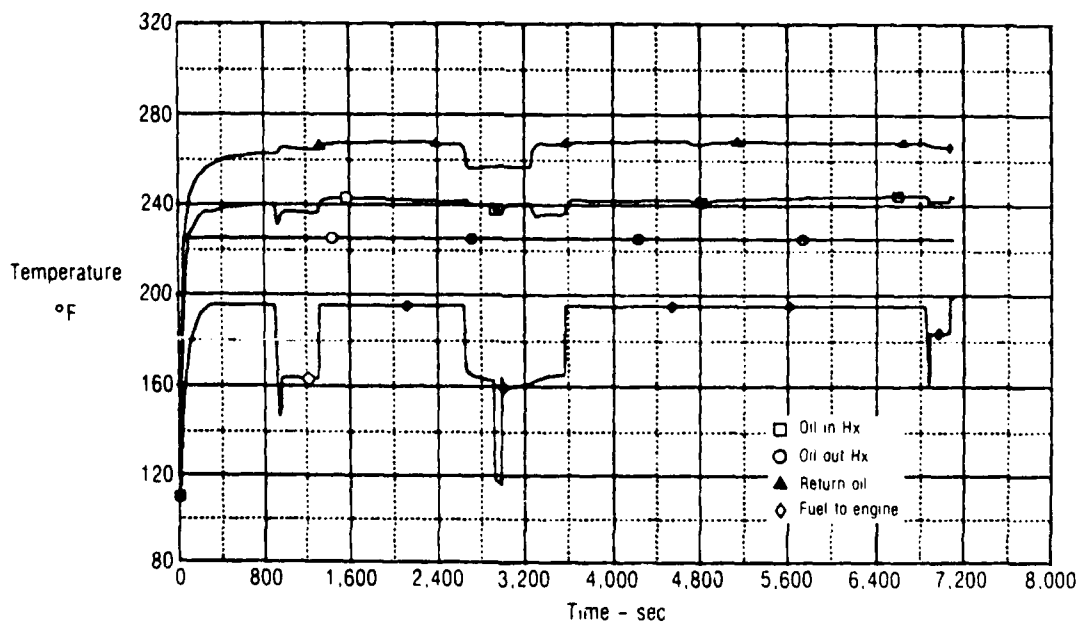


Figure 56
3,000 psi Baseline
PC-1 Ram Air HX Requirements

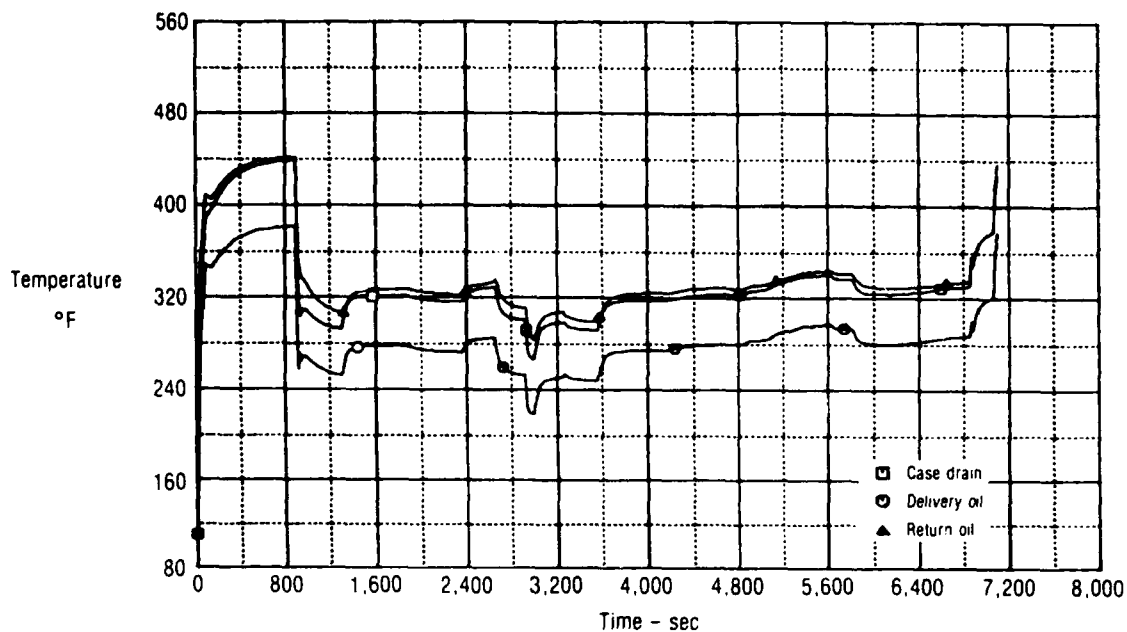


Figure 57
8,000 psi Baseline Without Ram Air HX
 Utility Pump and Return Temperatures

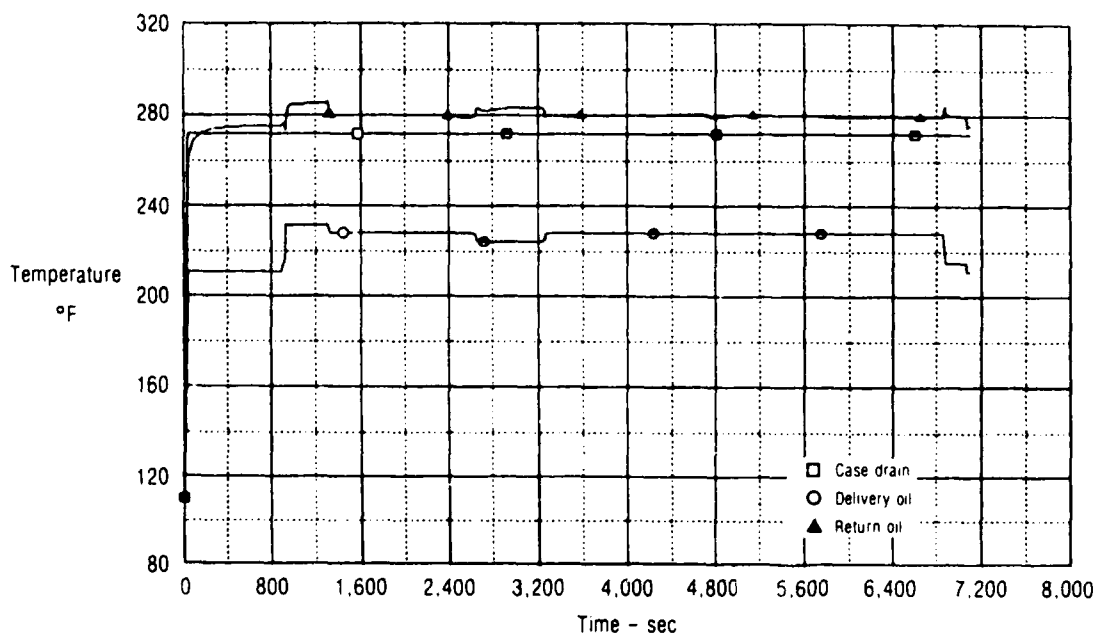


Figure 58
8,000 psi Baseline
 Utility Pump and Return Temperatures

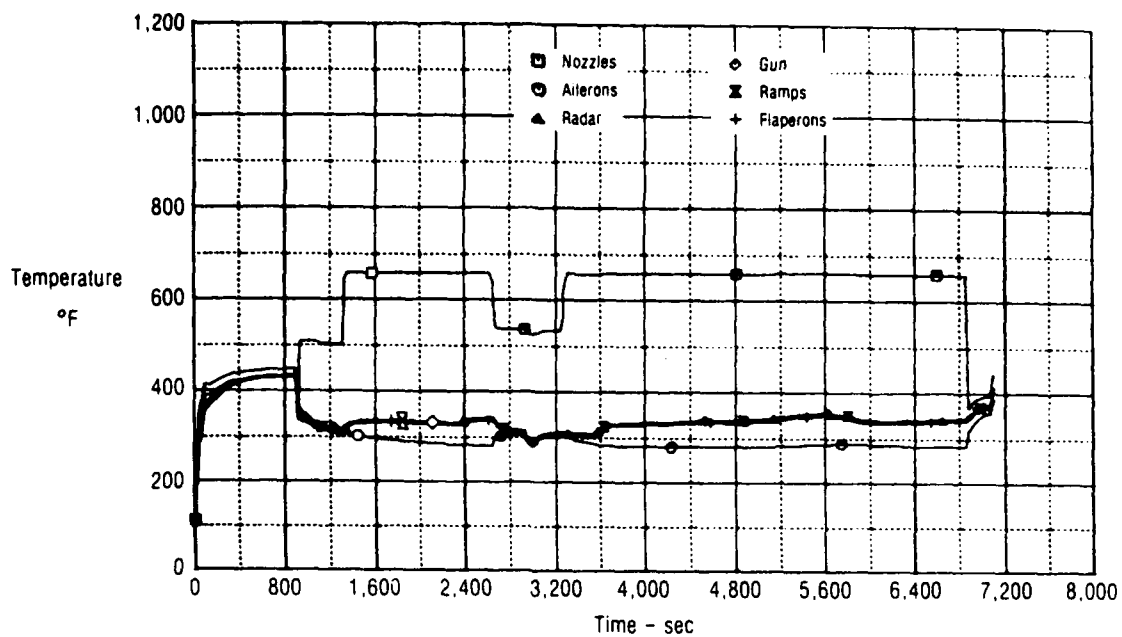


Figure 59
8,000 psi Baseline Without Ram Air HX
 Utility Actuator Exit Temperatures

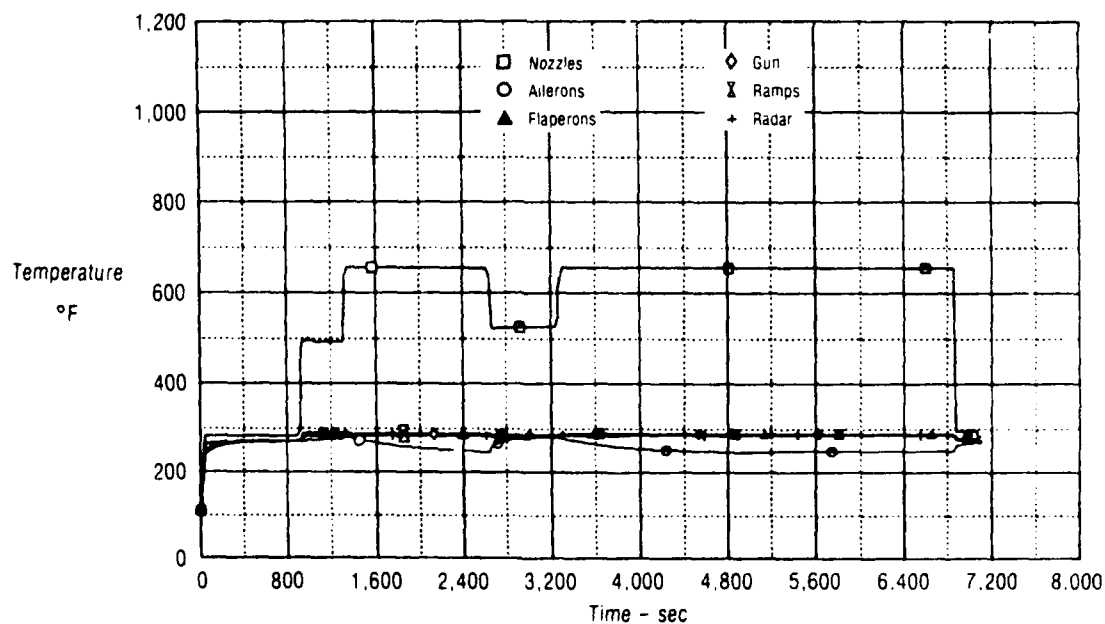


Figure 60
8,000 psi Baseline
 Utility Actuator Exit Temperatures

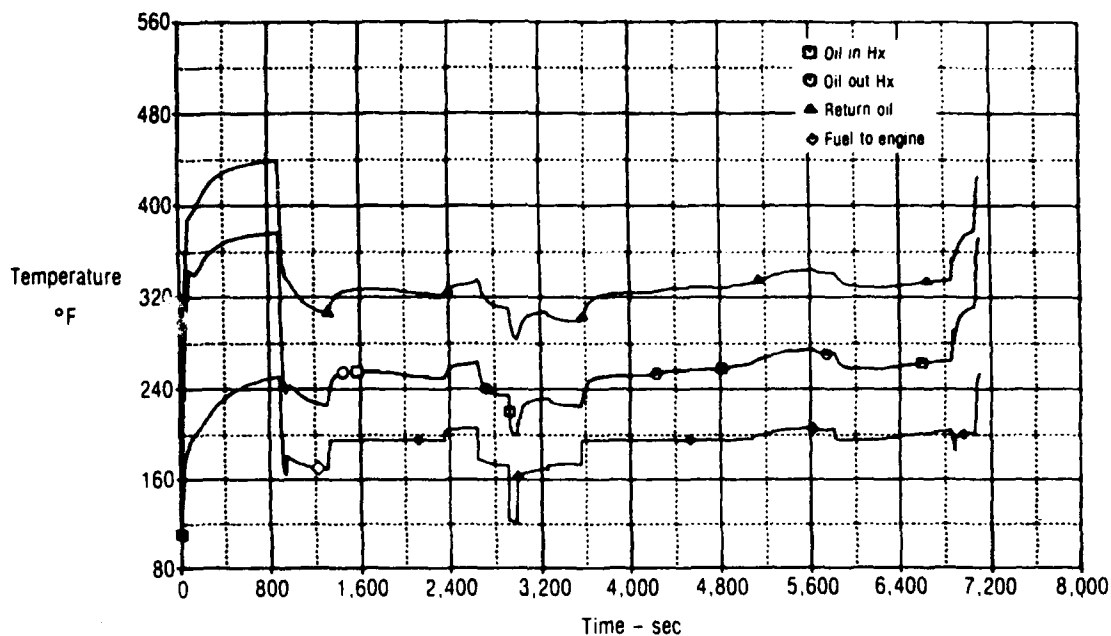


Figure 61
8,000 psi Baseline Without Ram Air HX
 Utility Ram Air HX Requirements

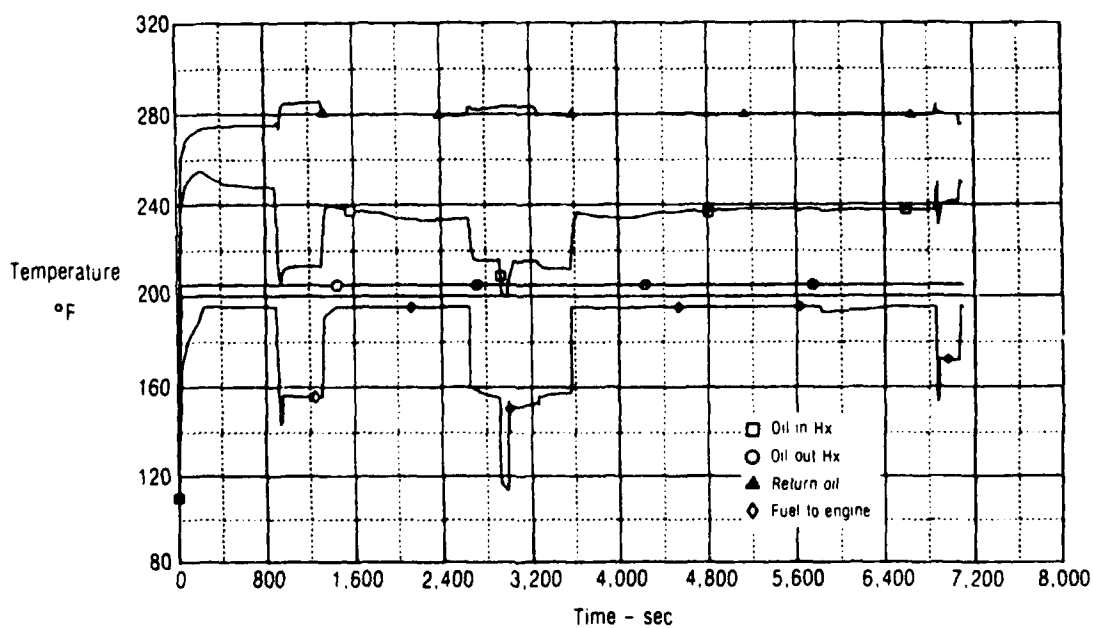


Figure 62
8,000 psi Baseline
 Utility Ram Air HX Requirements

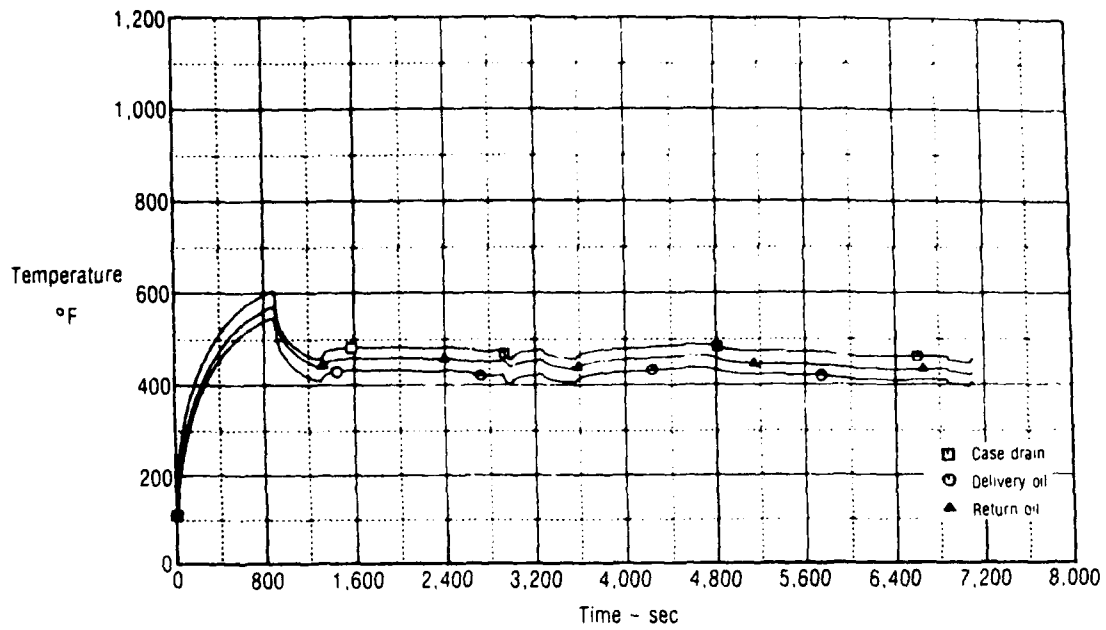


Figure 63
8,000 psi Baseline Without Ram Air HX
PC-1 Pump and Return Temperatures

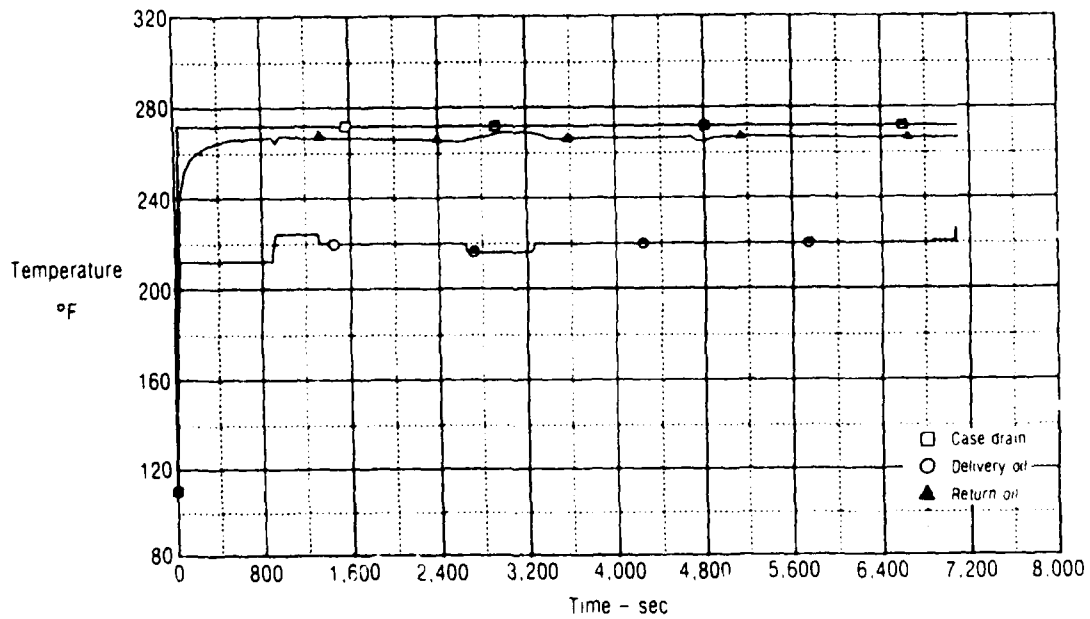


Figure 64
8,000 psi Baseline
PC-1 Pump and Return Temperatures

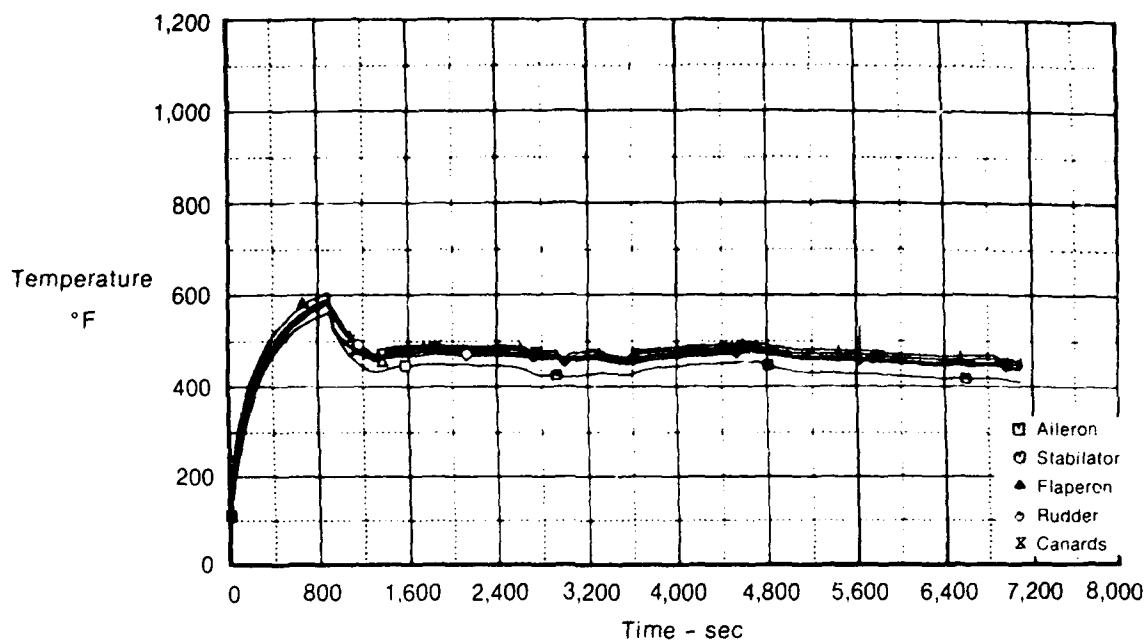


Figure 65
8,000 psi Baseline Without Ram Air HX
PC-1 Actuator Exit Temperatures

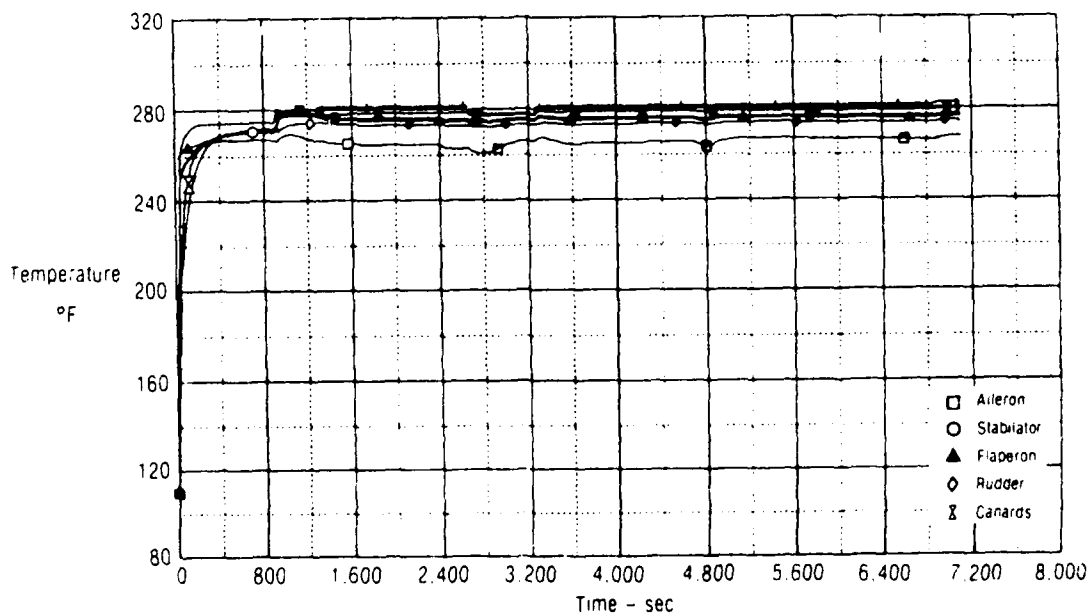


Figure 66
8,000 psi Baseline
PC-1 Actuator Exit Temperatures

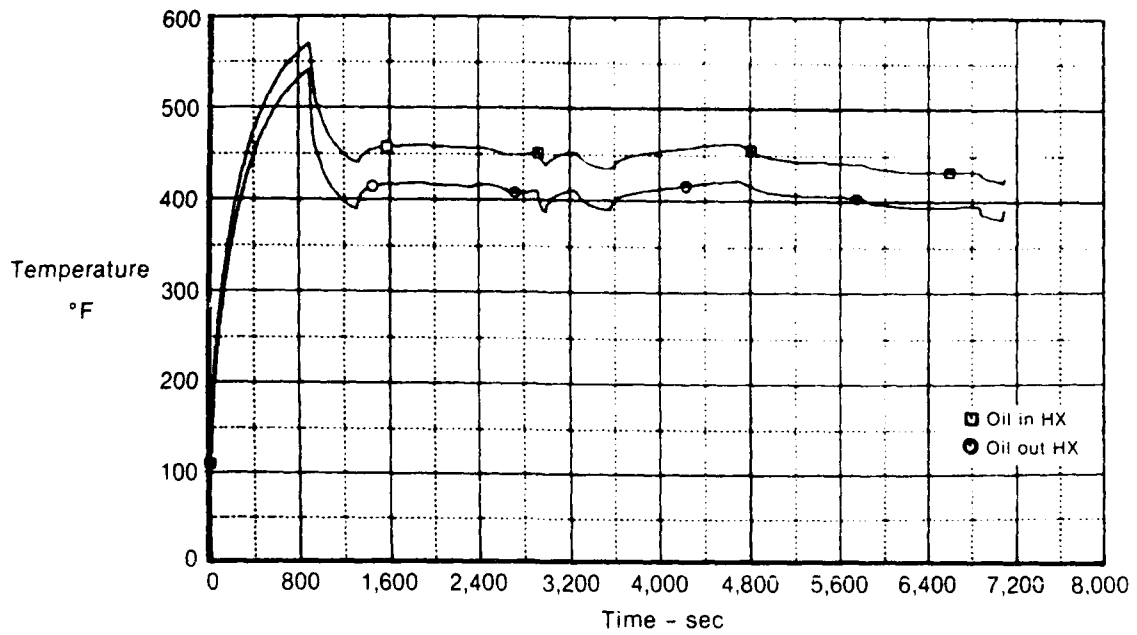


Figure 67
8,000 psi Baseline Without Ram Air HX
 PC-1 Ram Air HX Requirements

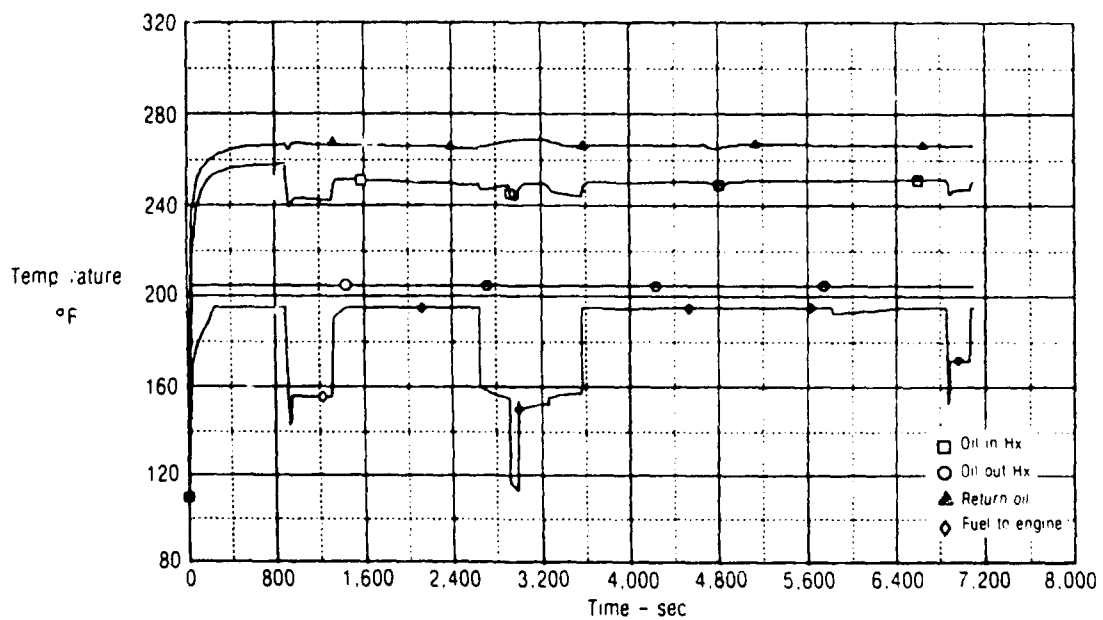


Figure 68
8,000 psi Baseline
 PC-1 Ram Air HX Requirements

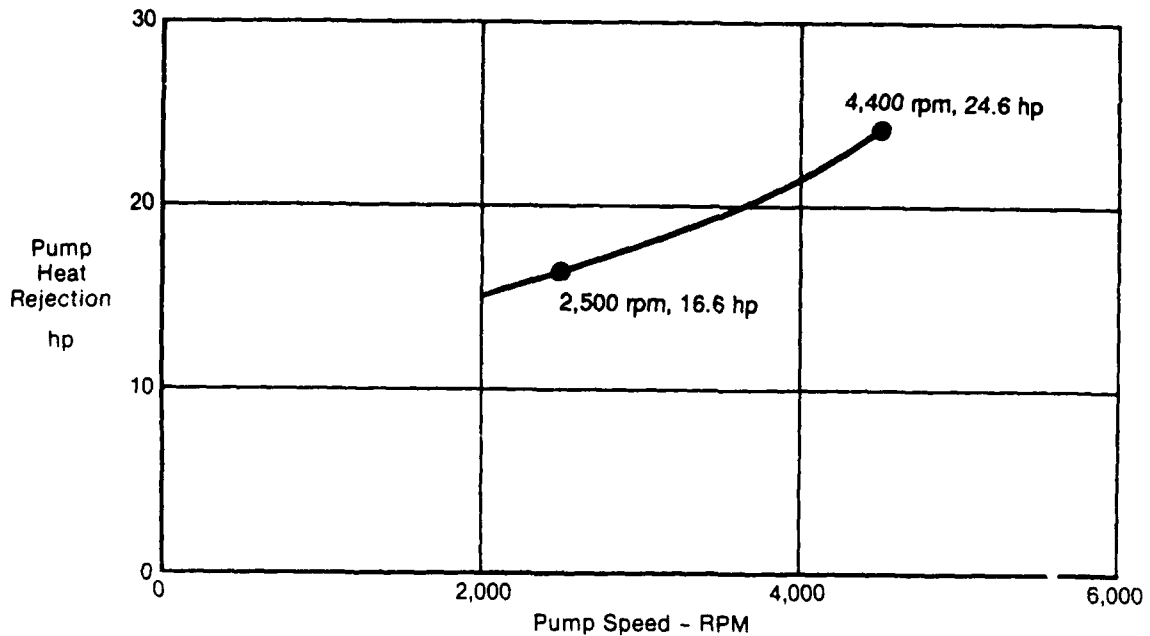


Figure 69
Pump Heat Rejection vs. Pump Speed
 2.30 In³/Rev 8,000 psi CTFE

The digital computer control system yielded an increase in null leakage due to digital noise. The 3,000 psi reference system null leakages did not incorporate digital noise, see Figure 70. Based on F-18 flight simulator test data, a valve that has a null leakage of 1.0 gpm with an analog computer, will increase 75 percent using a digital computer. This additional leakage flow is determined by three factors. The first factor is the noise created by the navigational sensors, i.e., gyros and accelerometers, that is sent to the flight control computer. The second is the noise generated when the digital computer is rounding off numbers to integer values. The third factor is the clear air turbulence that is inherent in all dynamic flight control simulations, which will create a dither on the ram of the control valve.

This problem can be alleviated by incorporating a 0.001-inch overlap on the valve spool which decreases null leakage to a level similar to that using an analog computer. The 0.001-inch overlap was used in determining the 8,000 psi null leakage flow rates shown in Figure 71.

Another factor included in the heat rejection analysis, was operational flow rates. The operational flow created a pressure drop in the distribution system, which is heat generation, and also created heat when the actuator was dynamically braking against an aiding external load. The operational flow rates used in the thermal analysis are shown in Figures 72 and 73.

Per Actuator	Null Leakage (per A/C)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	0.088	0.088	0.061	0.061	0.07	—	0.07	0.07	0.03	0.079
Flaperon	0.35	0.35	0.35	—	0.35	0.016	0.35	0.35	0.035	0.315
Rudder	0.053	0.053	0.037	0.047	0.047	—	0.047	0.047	—	—
Stabilator	0.70	0.70	0.07	0.35	0.56	0.14	0.56	0.42	0.07	0.63
Canard	0.70	0.70	0.175	0.49	0.63	0.56	0.63	0.49	0.35	0.63
Utility System										
Gun	0.61	0.61	0.61	0.61	0.61	—	0.61	0.61	0.61	0.61
Steering	0.08	0.02	0.02	—	—	—	—	—	—	0.02
Radar	—	—	—	—	—	—	—	—	—	—
Nozzles	2.80	2.80	—	—	—	—	—	1.40	2.54	2.80
Ramps	1.05	1.05	1.043	1.040	1.040	0.858	1.040	1.040	1.043	1.05
Aileron	0.175	0.175	0.123	0.123	0.14	—	0.14	0.14	0.058	0.158
Flaperon	0.70	0.70	0.70	—	0.70	0.70	0.70	0.70	0.70	0.70

*Flows are mean values over flight phase in GPM

Figure 70
F-15 SMTD
 Leakage Flow 3,000 Psi With Force Motors

Per Actuator	Null Leakage (per A/C)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	0.130	0.130	0.092	0.092	0.104	—	0.104	0.104	0.042	0.118
Flaperon	0.50	0.50	0.50	—	0.50	0.50	0.50	0.50	—	0.45
Rudder	0.10	0.10	0.10	0.07	0.09	—	0.09	0.09	—	—
Stabilator	1.00	1.00	0.10	0.50	0.80	0.20	0.80	0.60	0.10	0.90
Canard	1.00	1.00	0.25	0.70	0.90	0.80	0.90	0.70	0.50	0.90
Utility System										
Gun	0.34	0.34	0.34	0.34	0.34	—	0.34	0.34	0.34	0.34
Steering	0.11	0.032	0.032	—	—	—	—	—	—	0.032
Radar	—	—	—	—	—	—	—	—	—	—
Nozzles	4.16	4.16	—	—	—	—	—	2.08	3.786	4.16
Ramps	1.56	1.56	1.55	1.528	1.528	1.326	1.528	1.55	1.55	1.440
Aileron	0.26	0.26	0.182	0.208	—	—	—	0.208	0.086	0.234
Flaperon	1.00	1.00	1.00	—	1.00	1.00	1.00	1.00	—	0.90

*Flow rates mean values over flight phase in GPM

Figure 71
F-15 STMD
 Leakage Flow 8,000 Psi With Force Motors Without Digital Noise

Per Actuator	Actuator Flow (Unloaded)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	1.08	0.228 ¹	0.034	0.034	0.023	0.164	0.023	0.023	0.076	0.011
Flaperon	1.31	0.875 ¹	—	0.137	—	0.051	—	—	0.122	0.014
Rudder	1.37	0.175 ¹	0.043	0.015	0.015	0.208	0.015	0.015	0.143	0.143
Stabilator	11.60	1.75 ¹	1.092	0.605	0.242	1.732	0.242	0.484	1.092	0.121
Canard	9.81	1.75 ¹	0.768	0.307	0.102	1.489	0.102	0.307	0.512	0.102
Utility System										
Gun	10.88	—	—	—	—	0.107	—	—	—	—
Steering	0.90	0.64	0.64	—	—	—	—	—	—	0.64
Radar	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64
Nozzles	—	7.28	5.81	0.204	0.061	0.162	0.061	0.039	1.08	0.54
Ramps	5.91	—	0.006	0.008	0.008	0.213	0.008	0.006	0.006	—
Aileron	2.17	0.456 ¹	0.069	0.069	0.046	0.329	0.046	0.046	0.152	0.022
Flaperon	2.61	1.75 ¹	—	0.274	—	0.099	—	—	0.244	0.028

¹ = Leakage

*Flows are mean values over flight phase in GPM

Figure 72
F-15 SMTD
 Operational Flow 8,000 Psi With Force Motors Without Digital Noise

Per Actuator	Actuator Flow (Unloaded)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	2.89	0.088 ¹	0.091	0.091	0.06	0.439	0.06	0.06	0.202	0.030
Flaperon	3.48	0.35 ¹	—	0.363	—	0.132	—	—	0.326	0.037
Rudder	3.64	0.053 ¹	0.114	0.038	0.038	0.5523	0.038	0.038	0.380	0.380
Stabilator	30.92	0.70 ¹	2.91	1.612	0.645	4.69	0.645	1.29	2.91	0.322
Canard	26.15	0.70 ¹	2.046	0.818	0.272	3.97	0.272	0.818	1.364	0.272
Utility System										
Gun	29.0	—	—	—	—	0.284	—	—	—	—
Steering	2.41	1.71	1.71	—	—	—	—	—	—	1.71
Radar	1.70	1.70	1.70	1.70	1.70	1.70	1.70	1.70	1.70	1.70
Nozzles	—	—	16.0	0.544	0.162	0.431	0.162	0.103	2.88	1.44
Ramps	105.52	—	0.097	0.136	0.136	3.79	0.136	0.136	0.097	—
Aileron	5.78	0.010 ¹	0.180	0.180	0.121	0.877	0.121	0.121	0.404	0.06
Flaperon	6.96	0.042 ¹	—	0.726	—	0.264	—	—	0.653	0.073

¹ - Leakage

*Flows are mean values over flight phase in GPM

Figure 73
F-15 SMTD
 Operational Flow 3,000 Psi With Force Motors

The last major factor included in hydraulic system heat rejection was pump inefficiency. For this thermal analysis, it was assumed that two-thirds of the pump heat rejection was routed to the case drain, and the remaining one-third was output with the pump outlet flow.

Figures 74 and 75 depict the maximum simultaneous flow requirements for the 3,000 and 8,000 psi F-15 SMTD.

The values in the first column are the maximum simultaneous flow rates. This condition occurs during the combat phase of the mission profile with high levels of maneuverability. The maximum no-load flow rates were given in the specifications for the individual components. The PC pumps/systems were sized to provide an emergency backup to the engine nozzle actuators in the event of a utility hydraulic system failure.

The preliminary column shows a significant decrease in the air inlet control actuator maximum flow rates, therefore decreasing the maximum simultaneous flow rate. This reduction resulted from an analysis which indicated the high actuator rates on the production F-15 are not necessary to achieve the required performance on the F-15 SMTD.

	Utility System			PC System		
	No-Load 8,000 psi	Preliminary 8,000 psi	Final 8,000 psi	No-Load 8,000 psi	Preliminary 8,000 psi	Final 8,000 psi
First Ramp Actuator	17.34	3.47	3.47	Aileron Actuators	1.08	1.08
Diffuser Ramp Actuator	13.40	0.67	0.67	Flaperon Actuators	1.31	—
Bypass Door Actuator	8.83	1.77	1.77	Rudder Actuators	1.12	1.12
Nose Wheel Steering Unit	—	—	—	Stabilator Actuators	8.38	8.38
Heat Exchanger Door Actuator	—	—	—	Canard Actuators	7.40	7.40
Aerial Refuel Actuator	—	—	—	Backup for Nozzle Actuators	28.00	14.00
Radar Drive	0.64	0.64	0.64	Total	37.29	31.98
Gun Drive	10.88	10.88	10.88			25.98
Nozzle Actuators	28.00	28.00	16.00			
Aileron Actuators	2.17	2.17	2.17			
Flaperon Actuators	2.61	—	—			
Total	83.87	47.60	35.60			

Figure 74
Maximum Simultaneous Combat Flow Rates (GPM) at 8,000 psi

Utility System			
	No-Load 3,000 psi	Preliminary 3,000 psi	Final 3,000 psi
First Ramp Actuator	46.24	9.25	9.25
Diffuser Ramp Actuator	35.74	1.79	1.79
Bypass Door Actuator	23.54	4.71	4.71
Nose Wheel Steering Unit	—	—	—
Heat Exchanger Door Actuator	—	—	—
Aerial Refuel Actuator	—	—	—
Radar Drive	1.70	1.70	1.70
Gun Drive	29.00	29.00	29.00
Nozzle Actuators	60.00	60.00	34.30
Aileron Actuators	5.78	5.78	5.78
Flaperon Actuators	6.94	—	—
Total	208.94	112.23	86.93

PC System			
	No-Load 3,000 psi	Preliminary 3,000 psi	Final 3,000 psi
Aileron Actuators	2.89	2.89	2.89
Flaperon Actuators	3.48	—	—
Rudder Actuators	3.64	2.99	2.99
Stabilator Actuators	30.92	22.34	22.34
Canard Actuators	26.15	19.74	19.74
Backup for Nozzle Actuators	60.00	60.00	34.30
Total	127.08	107.96	82.26

Figure 75
Maximum Simultaneous Combat Flow Rates (GPM) at 3,000 psi

The final column shows another reduction in flow rates due to adding regenerative flow to the nozzle actuators. This resulted in a 42.8 percent decrease in nozzle actuator flow rate and a 25 percent decrease in pump capacity.

This reduction, shown in Figure 74, prompted an analysis of pump resizing. Initially, four 40 gpm pumps were required to achieve the simultaneous maximum flow requirements as per the 8,000 psi no-load requirement. As the final 8,000 psi column indicates, the flow requirement was reduced to the level where four 25 gpm pumps could be used, two for the utility system and one for each PC system.

It was decided to use the 40 gpm pump with reduced rotational speed. This decreased pump heat rejection as shown in Figure 76, and increased pump life and reliability, while adding only a minimal weight penalty.

At 8,000 psi, the pump heat rejection used in the thermal model was 16.64 horsepower, or 66.6 horsepower per aircraft.

	Compensated Flow	
	4,400 RPM	2,500 RPM
8,000 psi 2.30 in. ³ /Rev	24.4 hp	16.64 hp
	Compensated Flow	
	3,400 RPM	1,930 RPM
3,000 psi 8.62 in. ³ /Rev	23.25 hp	15.6 hp

Figure 76
F-15 SMTD Pump Heat Rejection

(3) Heat Exchanger/Environmental Control System (ECS) Weight - After the maximum temperature levels were determined, the additional heat exchanger requirement was assessed. The production F-15 aircraft was designed such that when the hydraulic oil delivery temperature increased to 225°F, the oil was bypassed to a fuel/oil heat exchanger to maintain a fluid temperature less than 275°F while maintaining aircraft fuel temperature levels less than 195°F. These criteria were used for the initial 8,000 psi thermal analysis. However, the additional heat loads of the SMTD configuration for both 3,000 and 8,000 psi, extended the fuel beyond its heat sink capacity. When the fuel/oil heat exchanger reached its saturation point, an additional ram air heat exchanger was incorporated to handle the heat load.

For the SMTD at 3,000 psi, the 225°F control temperature met the requirement goals of 195°F fuel and 275°F hydraulic oil maximum temperatures with the addition of ram air heat exchangers. Figure 77 shows the heat exchanger weight breakdown for the 3,000 psi baseline, 3,000 psi SMTD and 8,000 psi SMTD.

The total heat exchanger weight included ram air heat exchangers, ram circuit, installation hardware and fuel/oil heat exchanger.

It was decided that when using a digital flight control system, 0.001 valve overlap would be incorporated to reduce valve leakage. Another thermal analysis was performed because of the decrease in hydraulic system heat load.

The 8,000 psi system was analyzed in a different fashion to overcome the higher heat load. To achieve the goal of limiting the hydraulic oil temperature to 275°F and the fuel temperature to 195°F, the control temperature had to be set at 200°F vs. 225°F at 3,000 psi during the taxi and takeoff segments, and to 220°F for the remainder of the flight.

The heat rejection for the modified 3,000 psi production F-15 SMTD shown in Figure 78 was calculated. This configuration added engine nozzle actuator regeneration and 0.001 overlap valves. No thermal analysis was performed and no heat exchanger sizing was made.

	3,000 psi F-15 Production		3,000 psi SMTD Modified Production		8,000 psi SMTD Baseline Production	
	PC-1 and PC-2	Utility	PC-1 and PC-2	Utility	PC-1 and PC-2	Utility
Fuel Oil HX	6.96	6.96	9.1	9.1	9.1	9.1
Ram Air HX	—	—	2.9	4.3	13.4	13.7
Ram Circuit	—	—	4.0	3.8	9.0	7.5
Fans and Motor	—	—	18.6	20.4	51.8	44.1
Installation	0.39	6.45	6.4	7.1	18.6	16.3
Subtotal	7.35	13.41	41.0	44.7	101.9	90.7
Total Utility and PC - lb	20.76		85.7		192.6	
System Heat Rejection - hp	32.30		74.0		109.6	

Figure 77
Heat Exchanger Weight Breakdown

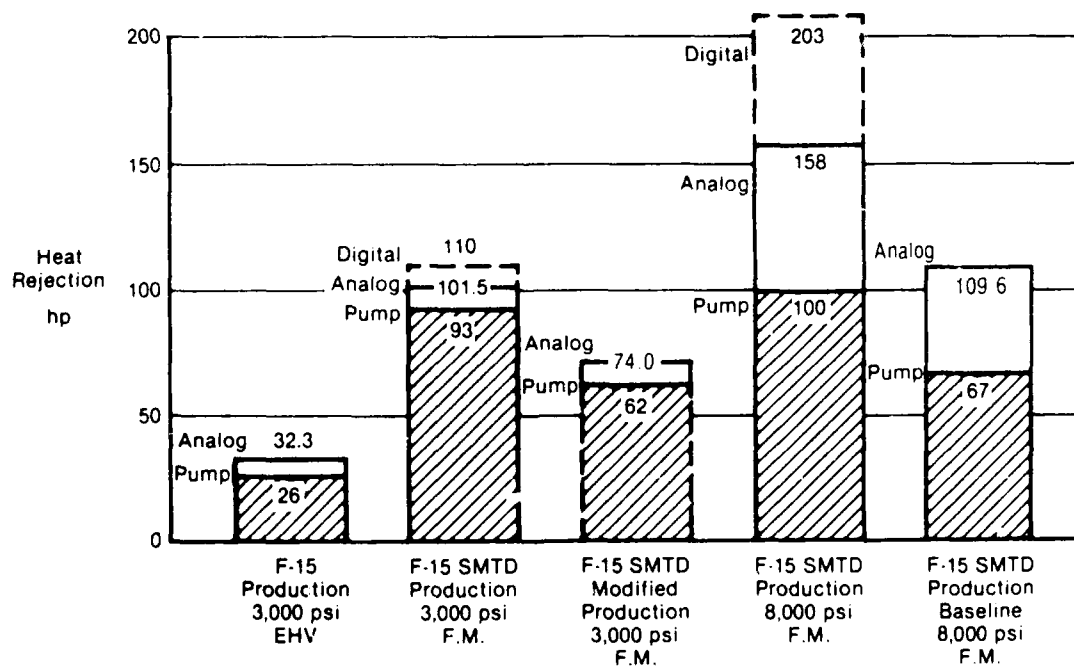


Figure 78
Heat Rejection by Aircraft Configuration

Potential locations for the ram air heat exchangers on the F-15 SMTD airplane are depicted in Figure 79.

c. Life Cycle Cost Analysis

(1) Analytical Approach - LCCs were developed on a subsystem basis (hydraulics and related equipment) and on a system basis (total aircraft) using the procedure shown in Figure 80.

The LCC of the hydraulic and related equipment was calculated using the RCA PRICE model. Development, production and support costs were estimated for the F-15. However, the results are generally applicable to other fighter aircraft of that weight class.

The LCC was calculated using a method that resized the aircraft to maintain constant performance for changes in hydraulic system weight. The F-15C aircraft with its 3,000 psi hydraulic subsystem was used as a reference.

The reference aircraft system was compared to a reference F-15 SMTD redesigned to a production aircraft configuration with a 3,000 psi hydraulic subsystem. This 3,000 psi production configuration was then redesigned to an 8,000 psi CTFE subsystem for the baseline LECHT F-15 SMTD aircraft system. LCCs were generated for each of these configurations.

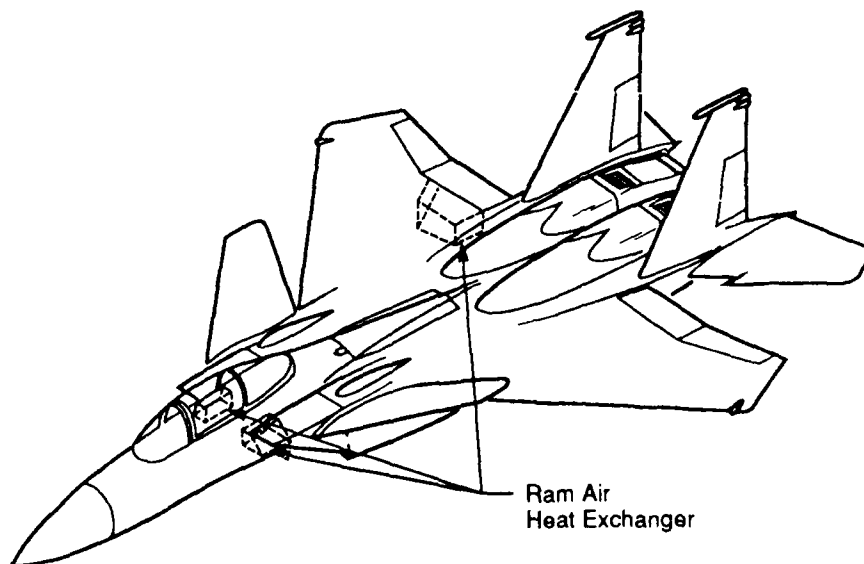


Figure 79
Heat Exchanger Locations
CTFE 8,000 psi SMTD Ram Air

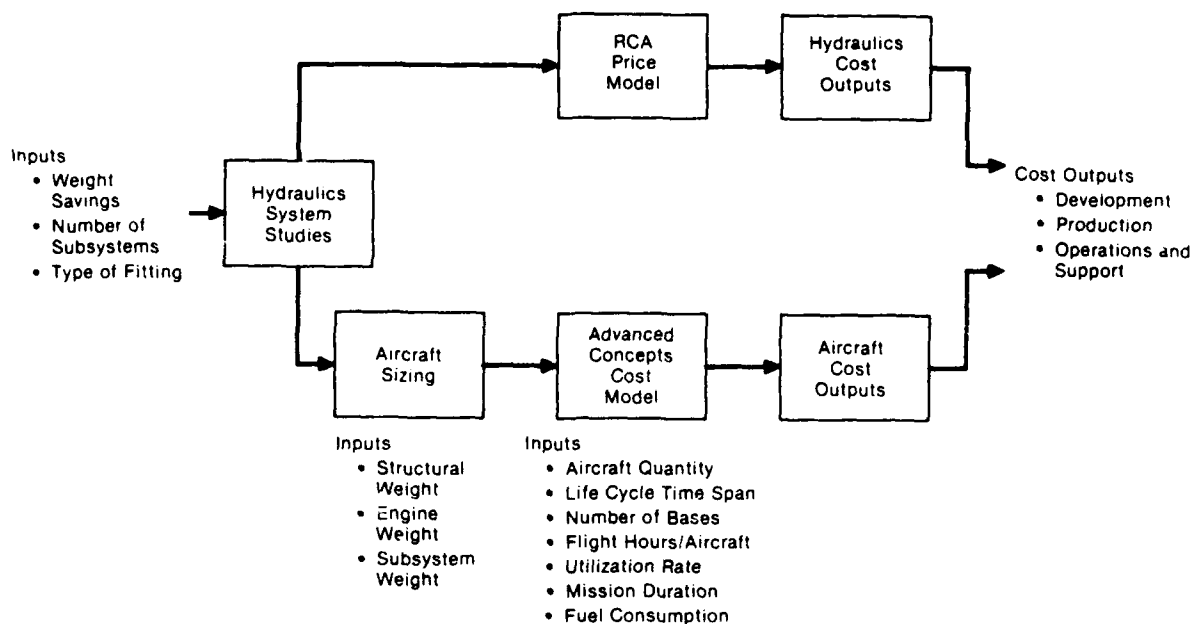


Figure 80
Cost Analysis Procedure

(2) Reliability and Maintainability - F-15 reliability and maintainability data was used from the USAF 66-1 maintenance data collection system. For the 3,000 psi and 8,000 psi hydraulic systems, reliability was considered the same. Figure 81 shows the reliability and maintainability assessment techniques and parameters used to develop data. The Mean Flight Hours Between Failures (MFHBF) and the Mean Time To Repair (MTTR) are shown in Figure 82.

- Techniques Utilized in F-15, F/A-18, AV-8B and Future Programs
- Like/Similar Evaluation of Field Data
 - AFM 66-1, Navy 3M, MCAIR Component Life Evaluation and Reliability (CLEAR)
 - Vendor Predictions
 - Logistics Support Analysis (LSA) Predictions Per MIL-STD-1388
- Reliability Develops MTBF
- Maintainability Develops MTBMA, MTTR, MMH/FH
 - All Facets of Integrated Logistics Support (ILS) Evaluated
 - Concepts Verified
 - Optimum Repair Level Analysis (ORLA), Level of Repair Analysis (LORA)
 - Modeling Logic Control Output Module (LCOM), Comprehensive Aircraft Support Effectiveness Evaluation (CASEE), Etc.)
- Data Input to Operating and Support (OPS) Analysis
 - Life Cycle Cost Derived

Figure 81
Reliability and Maintainability Assessment

Actuators				
Application	Type*	Quantity/ Aircraft	Reliability MFHBF	Maintainability MTTR
Flight Control				
Rudders	SWFM	2	1,844	6,187
Ailerons	SWFM	2	900	4,350
Flaperons	SWFM	2	900	4,350
Canards	SWFM	2	227	6,235
Stabilators	SWFM	2	227	6,235
Engine Nozzle				
Upper Vane	SWFM	2	838	5,000
Lower Vane	SWFM	2	838	5,000
Convergent Flap	SWFM	4	419	5,000
Outboard Flap	SWFM	4	419	5,000
Inboard Flap	SWFM	4	419	5,000
Utility				
Canopy Lock	Sim	1	7,915	5,953
Refuel	Sim	1	79,151	3,160
Arresting Hook	Sim	1	8,795	3,427
ECS	Sim	1	79,151	4,960
NLG Uplock	Sim	1	3,958	3,783
MLG Uplock	Sim	2	79,151	3,907
Canopy Main	Sim	1	1,365	3,408
Air Induction Bypass Door	SWFM	2	5,863	5,444
NLG Retract	Sim	1	5,654	5,000
MLG Retract	Sim	2	4,947	5,000
Air Induction First Ramp	SWFM	2	4,947	6,004
Air Induction Diffuser	SWFM	2	13,192	6,576
Speedbrake	Sim	1	5,654	3,196
Flight Control Switching				
Ailerons	Mech	2	2,553	5,893
Canards	Mech	2	4,059	5,893
Stabilators	Mech	2	4,059	5,000
Engine Nozzle				
Actuator Control	Mech	2	4,059	5,000

*Legend

SWFM = Servo With Force Motor
Sgl = Sinehard
MECH = Mechanical
Sim = Simple

Actuators				
Application	Type*	Quantity/ Aircraft	Reliability MFHBF	Maintainability MTTR
Utility				
Canopy	Sol	1	1,885	5,451
Refuel	Sol	1	19,788	3,520
Arresting Hook	Sol	1	11,307	6,000
ECS	Sol	1	39,576	6,250
NLG				
Check Valve	Sim	1	19,788	6,250
Linear Directional Control	Sol	1	79,151	7,700
Restrictor	Sim	2	158,302	5,000
MLG				
Restrictor	Sim	2	158,302	5,000
Linear Directional Control	Sol	1	11,307	6,828
Brake	Mech	1	13,192	2,708
Emergency Extend	Sol	1	79,151	3,000
Miscellaneous				
Emergency Generator	Sol	1	79,151	5,000
Gun System	Sol	1	79,151	6,367
Utility				
Temperature Regulator	Mech	1	3,827	5,348
Primary HX	Mech	1	2,827	5,000
Jet Fuel Starter	Sol	1	79,151	5,000
PC-1, PC-2				
Temperature Regulator	Mech	2	1,414	3,204
Utility				
Pump	—	2	8,795	5,000
Reservoir	—	1	1,439	5,000
Primary HX	—	3	8,333	5,000
HX Fan	—	3	2,000	5,000
Canopy				
Accumulator	—	1	11,307	3,649
Jet Fuel Starter				
Hand Pump	—	1	3,598	4,170
Accumulator	—	1	682	5,149
PC-1, PC-2				
Pump	—	2	8,795	5,392
Reservoir	—	2	3,958	5,000

Figure 82
Reliability and Maintainability
MFHBF and MTTR 8,000 psi Baseline System Data

(3) Ground Rules and Assumptions - The LCCs consist of costs associated with full scale development, production, and 15 years of operations and support in the user environment. The ground rules are shown below:

- o Constant 1985 dollars
- o 20 Research Development Test and Evaluation Aircraft
- o 500 shipsets of hardware costed
- o Support equipment not costed
- o 15 year operational life
- o 300 flying hours per aircraft per year
- o Operational deployment in three theatres
- o Seven base-intermediate maintenance locations

(4) RCA PRICE Model Analysis - The hydraulic subsystem weights used in these estimates are shown in Figure 83. The miscellaneous components contain all valves, pumps, reservoirs and related attaching structures. The distribution system contains the tubing, fittings, clamps, brackets, etc.

Cost Category	3,000 psi		8,000 psi SMTD Baseline Production
	F-15 Production	SMTD Modified Production	
Flight Control Actuators	221.00	406.00	325.73
Engine Nozzle Actuators	—	300.00	284.36
Utility Actuators	207.00	147.30	115.40
Heat Exchangers	—	85.70	192.60
Miscellaneous Components	600.50	450.00	436.25
Distribution System	220.00	374.80	197.92
Fluid	163.00	219.19	181.77
Total	1,411.50	1,982.99	1,734.03

Note: (1) Weights are in pounds for one shipset of equipment

Figure 83
Hydraulic Subsystem Equipment Weight

The cost analysis for each equipment level shown was conducted at an individual component level, i.e., aileron actuator, rudder actuator, switching valve, utility pump, utility reservoir, and accumulated to the level shown in Figure 84. Each individual component is described in terms of independent model inputs.

LCC values are shown as delta increments for the production F-15 aircraft.

The 8,000 psi F-15 SMTD subsystem baseline was figured to be at a higher cost than the 3,000 psi F-15 SMTD, because component costs were higher. Adjusted LCC were calculated by translating the configuration weight increases into aircraft fuel costs. The 8,000 psi subsystem remained the highest cost.

	3,000 psi		8,000 psi SMTD Baseline Production
	F-15 Production	SMTD Modified Production	
Development	—	23	45
Procurement	—	106	176
Equipment	—	159	228
Initial Spares	—	- 5C	- 49
Other	—	- 3	- 3
Support	—	43	47
Replacement Spares	—	44	49
Maintenance Manpower	—	- 8	- 7
Other	—	7	5
Total Life Cycle Cost	—	172	268
Aircraft Fuel Costs	—	118	52
Total Life Cycle Cost (Adjusted)	—	290	320

Notes:

- (1) Equipment cost is based on purchase of 500 shipsets plus spares.
- (2) Fuel savings are for 15 years of operations and are based on total hydraulic and related weight savings.

Figure 84
Hydraulic and Related Equipment
ΔCost - Millions of FY 85 Dollars

A comparison of unit production hydraulic subsystem costs is presented in Figure 85. The 8,000 psi F-15 SMTD baseline subsystem has the highest unit cost, because component costs are higher. However, as shown in the following sections, the total aircraft LCC is lower because the total aircraft weight is lower.

(5) Advanced Concept Cost Model (ACCM) Analysis - Data presented for the production 3,000 psi F-15 reference subsystem, is representative of the F-15C. However, data for the 8,000 psi subsystem assumed a resized aircraft and not a retrofit of existing aircraft. The weight of the 3,000 psi production F-15 SMTD reference and 8,000 psi F-15 SMTD baseline hydraulic subsystems, were extrapolated from the F-15 production 3,000 psi subsystem.

After the weight impact was determined for the 3,000 psi and 8,000 psi F-15 SMTD hydraulic subsystems, the aircraft was resized by growth factor analysis to determine the total aircraft weight. This weight was then distributed to structure, fuel, engines and subsystems. The resulting weight breakdowns are displayed in Figure 86. Figure 87 shows the Life Cycle Costs summary.

Cost Category	3,000 psi		8,000 psi SMTD Baseline Production
	F-15 Production	SMTD Modified Production	
Flight Control Actuators	—	187	216
Engine Nozzle Actuators	—	231	286
Utility Actuators	—	- 27	- 9
Heat Exchangers	—	44	91
Miscellaneous Components	—	- 125	- 100
Distribution System	—	65	20
Fluid	—	0.2	1.8
Flight Control Computer	70	—	—
Integration and Test	—	16	24
Total	—	321.2	459.8

Figure 85
ΔUnit (Shipset) Procurement Cost Deltas
Thousands of FY 85 Dollars

Weight Group	3,000 psi F-15 Production		3,000 psi SMTD Modified Production				8,000 psi SMTD Baseline Production			
	Total Weight	Hydraulic Contribution	Total Weight	Hydraulic Contribution	Δ Weight	ΔHydraulic Contribution	Total Weight	Hydraulic Contribution	Δ Weight	ΔHydraulic Contribution
Airframe	13,718	140.70	14,438	147.82	+ 720	+ 7.12	14,123	144.85	+ 405	+ 4.15
Engine	6,061	—	6,321	—	+ 260	—	6,208	—	+ 147	—
Avionics	1,845	24.90	1,862	—	+ 17	- 24.90	1,855	—	+ 10	- 24.90
Subsystems	5,759	1,270.40	5,939	1,816.97	+ 180	+ 546.17	5,861	1,589.18	+ 102	+ 318.38
Total Empty Weight	27,383	1,436.40	28,560	1,964.79	+ 1,177	+ 528.39	28,047	1,734.03	+ 664	+ 297.63
Fuel	13,455	—	13,925	—	+ 470	—	13,728	—	+ 273	—
Payload	2,040	—	2,040	—	—	—	2,040	—	—	—
Oxygen	28	—	28	—	—	—	28	—	—	—
Crew	215	—	215	—	—	—	215	—	—	—
Unusable Fuel	493	—	493	—	—	—	493	—	—	—
Oil	76	—	76	—	—	—	76	—	—	—
Gun/Ammo	783	—	783	—	—	—	783	—	—	—
Misc Equipment	50	—	50	—	—	—	50	—	—	—
Gross Weight	44,523	1,436.40	46,170	1,964.79	+ 1,647	+ 528.39	45,460	1,734.03	+ 937	+ 297.63

Figure 86
Aircraft System Weight
Weight in Pounds

	3,000 psi F-15 Production		3,000 psi SMTD Modified Production		8,000 psi SMTD Baseline Production	
	Hydraulic	Total	Hydraulic	Total	Hydraulic	Total
Acquisition						
Development	—	—	23	130	45	139
Investment	—	—	106	502	176	539
Flyaway	—	—	159	443	228	487
Other	—	—	- 53	59	- 52	52
Subtotal	—	—	129	632	221	678
15 Years Operations and Support	—	—	43	386	47	279
Fuel	—	—	—	117	—	66
Total LCC	—	—	172	1,135	268	1,023
Unit Flyaway	—	—	0.321	0.887	0.461	0.974

Figure 87
Aircraft Life Cycle Cost Detail
 Δ Cost - Millions of FY 85 Dollars

(6) Hydraulic Subsystem and Aircraft Cost Summary - The total LCC of each hydraulic subsystem is presented in Figure 88. Both the 3,000 psi F-15 SMTD reference and 8,000 psi F-15 SMTD subsystems exhibit higher LCC than the 3,000 psi F-15C reference subsystem, primarily because the F-15 SMTD subsystems require additional equipment, i.e., engine nozzle and canard actuators, and heat exchangers. However, the 8,000 psi SMTD subsystem has a higher LCC than the 3,000 psi SMTD subsystem, which is the result of higher component costs.

Item	3,000 psi		8,000 psi SMTD Baseline Production
	F-15 Production	SMTD Modified Production	
Unit Flyaway Cost ⁽¹⁾	—	0.321	0.461
Life Cycle Cost ⁽²⁾	—	172	268
Weight (lb)	1,436.40	1,982.99	1,734.03

Notes:

- (1) Unit flyaway cost based on a purchase of 500 shipsets plus spares and does not include distribution system and fluid.
(2) Fuel costs not included.

Figure 88
Hydraulic Subsystem Cost/Weight Summary
 Δ Cost - Millions of FY 85 Dollars

The total cost and weight impact of each subsystem on an aircraft system, was determined by calculating LCC for each configuration after resizing the aircraft. The costs and weight for the total aircraft system, presented in Figure 89, show that the 8,000 psi F-15 SMTD baseline aircraft is less costly and lighter in weight than the 3,000 psi production F-15 SMTD reference aircraft.

Cost Category	3,000 psi		8,000 psi SMTD Baseline Production
	F-15 Production	SMTD Modified Production	
Total Aircraft System			
Unit Flyaway	—	0.887	0.974
Life Cycle Cost	—	1,135	1,023
Total Aircraft Empty Weight	27,383	28,560	28.047

Figure 89
Aircraft System Cost/Weight Summary
ΔCost - Millions of FY 85 Dollars

2.2 BASELINE CONFIGURATION AND EVALUATION CRITERIA (FOR PHASE II TRADEOFF STUDIES)

2.2.1 Baseline Configuration - The F-15 SMTD aircraft with CTFE hydraulic fluid at 8,000 psi using direct drive valves, was established as the baseline hydraulic system configuration. The baseline hydraulic system weight, thermal and LCC models were established for comparison.

The baseline hydraulic system weight, Figure 90, was divided into seven categories. As each energy savings concept was evaluated in Phase II, the effect on each category was apparent.

Thermal baseline models were established for the Utility and PC hydraulic systems. A typical mission profile was established to provide the baseline heat rejection. This same mission profile was used to provide the total hydraulic heat generation as each concept was evaluated in Phase II.

The thermal evaluation was used to size the baseline ram air heat exchangers for the heat exchanger weight category, see Figure 90. The total hydraulic system weight was then input into the LCC model.

Equipment	8,000 psi SMTD Baseline Production
Flight Control Actuators	325.73
Engine Nozzle Actuators	284.36
Utility Actuators	115.40
Heat Exchangers	192.60
Miscellaneous Components	436.25
Distribution System	197.92
Fluid	181.77
Total	1,734.03

Figure 90
Baseline Hydraulic Subsystem Weight

Three baseline parameters were established from the LCC analysis for comparison. These are (1) total aircraft system LCC, (2) total hydraulic subsystem LCC and (3) total aircraft empty weight, see Figure 91. The total aircraft weight was considered a baseline parameter since one pound of hydraulic system weight savings effected a three pound weight savings on the total aircraft.

Cost Category	8,000 psi SMTD Baseline Production
Total Aircraft System	
Unit Flyaway	0.974
Life Cycle Cost	1,023
Total Hydraulic Subsystem	
Unit Flyaway	0.460
Life Cycle Cost	268
Total Aircraft Empty Weight	28,047

Figure 91
Baseline Aircraft System Cost/Weight Summary
ΔCost - Millions of FY 85 Dollars

2.2.2 Evaluation Criteria for Phase II Tradeoff Studies - The impact of each concept on the total hydraulic system heat rejection can readily be evaluated individually. However, when the concepts are used in combinations, it may not be apparent which concepts provide the higher energy savings, or the order in which each concept should be incorporated to optimize the total heat rejection for the next concept. Also, weight and LCC have to be considered in conjunction with energy savings. Weighting factors were established for each parameter during Phase II, using an initial trial case.

Each concept were matched to a component, see Figure 92. The component was sized and its performance evaluated. Reliability and Maintainability evaluations established the MFHBF and MTTR for each component. Thermal evaluations defined the heat exchanger weight impact, which in turn impacted hydraulic system weight. Using these inputs, LCC analyses provided the total aircraft weight and costs.

Parameters established for concept evaluation were:

- o Hydraulic System Weight
- o Hydraulic System LCC
- o Hydraulic System Heat Load
- o Total Aircraft Weight
- o Total Aircraft LCC

2.3 CANDIDATE CONCEPTS

The candidate concepts identified for consideration in Phase II are summarized below:

- o Multipressure system pumps
- o Dry pump system pumps
- o Overlap valves
- o Flow augmentation
- o Load recovery valves
- o Flow augmented cooling flows
- o Pressure transformers
- o Servo hydraulic motors
- o Variable usage/displacement actuators
- o Optimized system/subsystem applications

Application of the candidate concepts required that an order be established to achieve a merit of performance for each concept and a merit of performance for combinations of the concepts. Phase II evaluated these concepts. However, the task was limited to a logical progression rather than a statistical evaluation of all combinations and permutations of the concepts. This logical progression was established by discussing the concepts in terms of pros, cons and ripple effects.

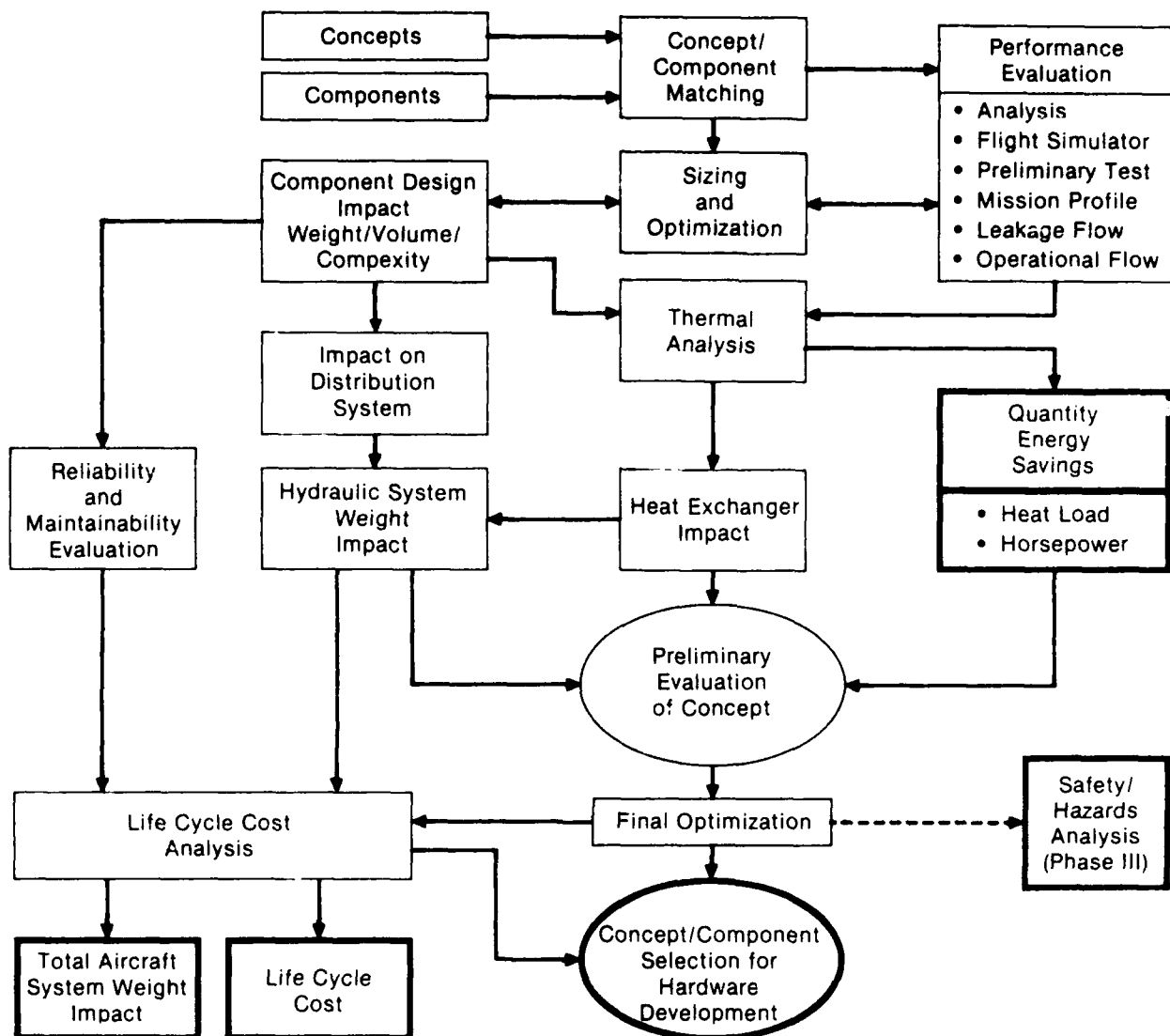


Figure 92
LECHT Concepts Evaluation
Flow Diagram

2.3.1 Intelligent Pumps - The concept of the variable pressure/variable flow, computer controlled pump was proposed on the basis that the flight controls and hydraulic utility functions did not require full system operating pressure at all times, as shown by a MCAIR study of an F-18 mission profile with flight test data shown in Figure 93. Although pump complexity was increased, this was outweighed by reduced heat rejection. The complexity was increased because of the added interface with the flight control actuators, see Figure 94.

Mission Segments	Cruise		Maneuvering			
	2,022	265	124.4	15.9	9.2	3.5
Flight Mission Time - sec	2,022	265	124.4	15.9	9.2	3.5
Percent of Mission Time	82.87	10.86	5.10	0.65	0.38	0.14
Average Hydraulic* Flow - GPM	9.3	10.3	15.2	24.8	35.1	43.7
GPM - Limits			< 20 30 40 >			
Hydraulic Efficient Applications	1,000 psi Intelligent Pump Effective 93.7% of the ACM Mission Time		1,000 psi - 3,000 psi Intelligent Pump			

*5 - 7 GPM leakage flow is included

Figure 93
F/A-18A Hydraulic/Flight Control System
Aircraft Combat Mission (ACM) Profile
 Recorded ACM Mission (Simulated Combat)
 Flight Time = 2,440 Sec at Yuma, Arizona With a
 Marine Pilot and a Data Sample Rate of 10 Hz

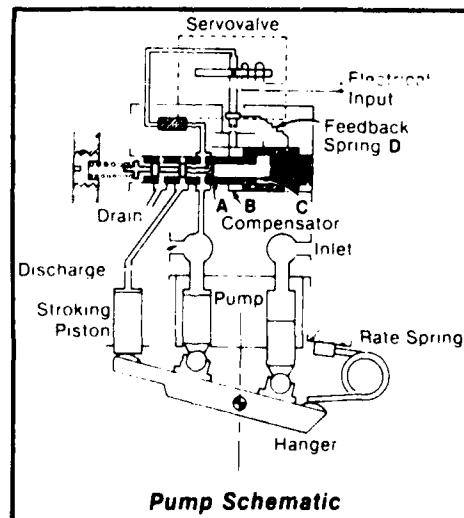


Figure 94
Intelligent Pump

Intelligent Pump

Advantages

- o Increased seal life
- o Reduced heat rejection
- o Allowed independent HP limiting operating modes
- o Subsystem which are not greatly enhanced by high pressure may be left as is (brakes)

Disadvantages

- o Non-optimum for load holding actuators, requiring pressure intensification.
- o Lowered reliability
- o Complicated switching valve/pressure sensing devices/monitors.
- o More difficult to troubleshoot (but more adaptive to automated maintenance concepts)
- o Two system components must allow for one system at high pressure and one at low pressure
- o Increased cumulative fatigue damage

2.3.2 Overlapped Valves - Valve overlap, illustrated in Figure 95, will reduce hydraulic power requirements by reducing the servovalve leakage which becomes significant when all servoactuators are considered, as shown in Figure 96.

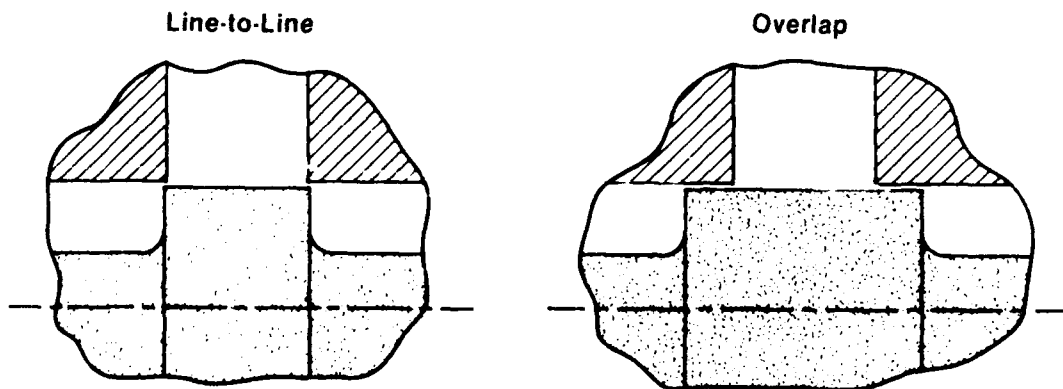


Figure 95
Overlap Valve Concept

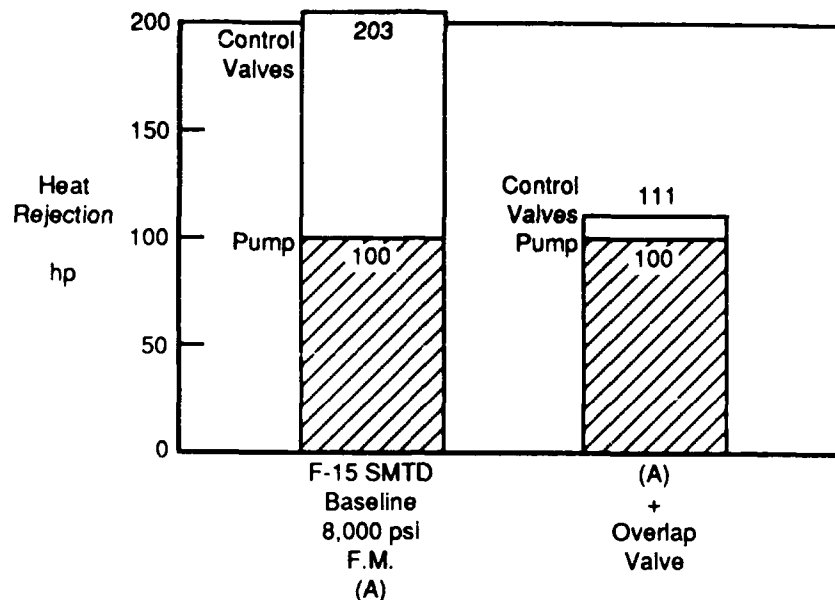


Figure 96
Overlap Valve Effects on Hydraulic Heat Rejection

Overlapped Valves

Advantages

- o Reduced leakage, i.e. steady state horsepower
- o No increased complexity
- o Negated flight control digital noise
- o No degradation in reliability
- o Negligible weight impact

Disadvantages

- o Potential deadband with consequential loss in performance

2.3.3 Flow Augmentation/Load Recovery Valves - These concepts reduced the pump flow demand when the actuator was operating at low load/no load/assisting load rates.

The flow augmentation concept used a jet pump to recirculate return flow to the pressure side when low load operation occurred, see Figure 97. The low flow demand on the pump reduced heat rejection and allowed for a smaller pump and supply lines. The projected reduction in pump heat rejection is shown in Figure 98.

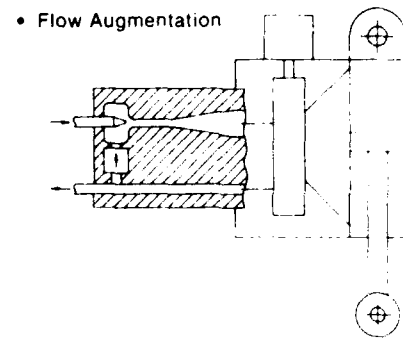
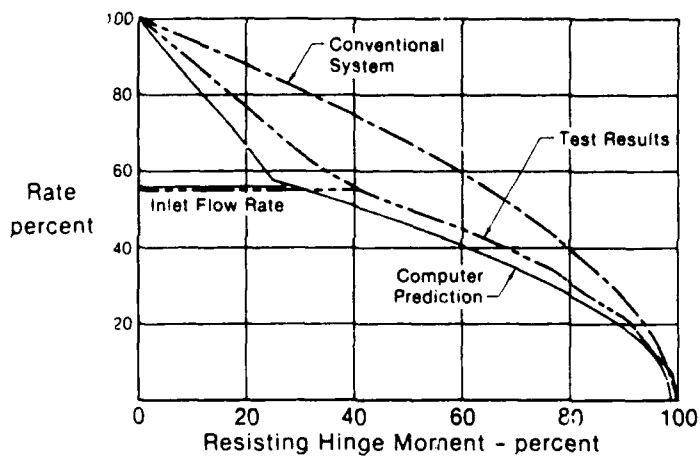


Figure 97
Flow Augmentation

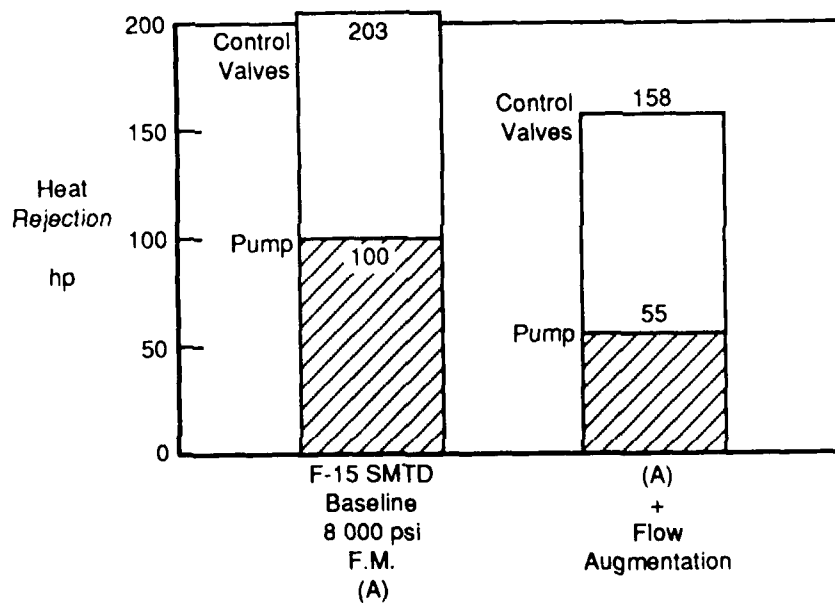


Figure 98
Flow Augmentation Effects on Hydraulic Heat Rejection

Flow Augmentation/Load Recovery Valve

Advantages

- o Greatly reduced peak flow demand to allow for smaller pump
- o No moving parts
- o Reduced distribution system size

Disadvantages

- o Complicated by variable pressure pump
- o Increased complexity
- o Required oversized actuators for some applications
- o Single point design

Load recovery valves are check valves that allow fluid to short circuit from the return side of the piston to the pressure side when an assisting load causes the pressure side to drop below the actuator outlet pressure, see Figure 99.

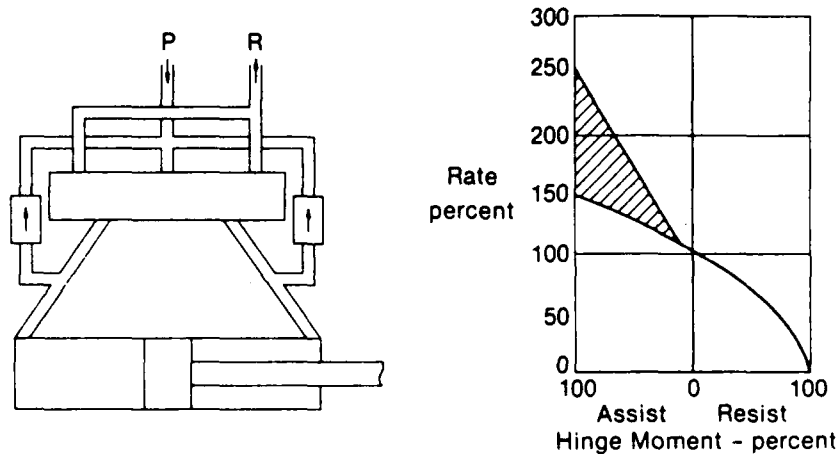


Figure 99
Load Recovery Valves

2.3.4 Pressure Intensifiers - Pressure intensification is an approach to providing high load holding capability with higher than normal pressure at the sacrifice of flow rate. The concept, (Figure 100), is applied to the system in conjunction with the intelligent pressure pump and in some instances, the flow augmentation device.

Pressure Intensification

Advantages

- o Allowed smaller area actuators in multipressure system

Disadvantages

- o Increased complexity and weight
- o Reduced reliability

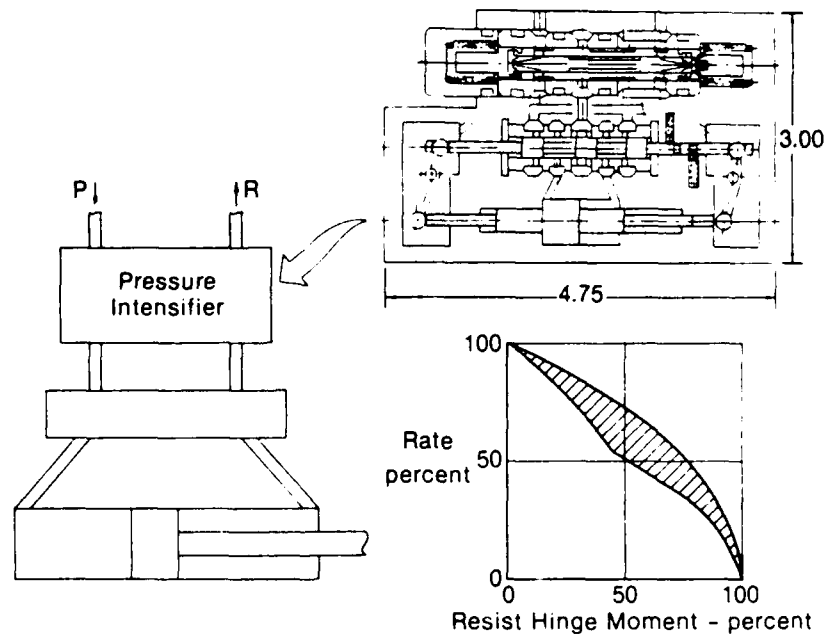


Figure 100
Pressure Intensifiers

2.3.5 Flow Augmented Cooling - This concept may be used in the engine nozzle actuators to pull hydraulic oil from the aircraft hydraulic system return for cooling flow. This oil is directed to cool the actuator LVDTs and the rod seals. To ensure system temperature is maintained within the fluid's limits, this concept may be the most practical method.

Flow Augmented Cooling

Advantages

- o Reduced pump demand
- o More efficient cooling of actuators
- o Increased actuator seal life
- o Increased electrical component life

Disadvantages

- o Increased complexity
- o Single point operating design
- o Increased system weight

2.3.6 Dry Sump System Pumps - This concept is based on the premise that the windage heat generation (power loss) associated with running the pump rotating group and frame wet in a full case, can be eliminated by scavenging the oil from the case with a small positive displacement pump.

Dry Sump Pump

Advantages

- o Lowered heat rejection
- o Increased efficiency
- o Returns approximately 25 cubic inches of oil per pump to the reservoirs. (i.e. smaller reservoirs)

Disadvantages

- o Increased pump complexity
- o Decreased reliability
- o Locally creates free air in system

The dry sump concept has no effect on anything but the pumps, motors and reservoir, and thereby can be evaluated alone. With proper attention to minimizing the disadvantages, this concept is projected to be implemented in future design, but requires quantification of the lower heat rejection and higher efficiency. It remains to be shown that the pump's power to scavenge is less than the windage loss eliminated.

2.3.7 Servohydraulic Motors - The servohydraulic motor considered was a variable displacement motor, whose displacement was controlled to achieve maximum efficiency and constant horsepower extraction or flow limiting. This concept can be evaluated on its own merits and affects the pump size and the system heat rejection.

Servohydraulic Motors

Advantages

- o Limits flow or power extraction
- o Allowed outside control of pump speed
- o Decreased pump size
- o Decreased heat rejection

Disadvantages

- o Increased weight
- o Increased complexity

2.3.8 Variable Usage/Displacement Actuators - The variable displacement actuator was based on dual tandem actuator configurations where extend and retract areas were selected to allow force staged operation to conserve power. The concept was to select the operating chambers and the operating system to achieve the lowest acceptable power extractions.

2.3.9 Optimized System/Subsystem Applications - Many utility functions may not benefit from the high system pressure as much as the flight controls. For this reason, it may be advantageous to operate these utility functions at lower pressures. Of primary interest is the Jet Fuel Starter (JFS) subsystem and accumulator, the canopy subsystem and the arresting hook. Also worthy of consideration, is the speedbrake subsystem, where high holding loads may be required at times when the multipressure pump is operating at low pressure.

SECTION III

PHASE II - TRADEOFF STUDIES

Phase II consisted of the analytical evaluation of several concepts and the following tasks:

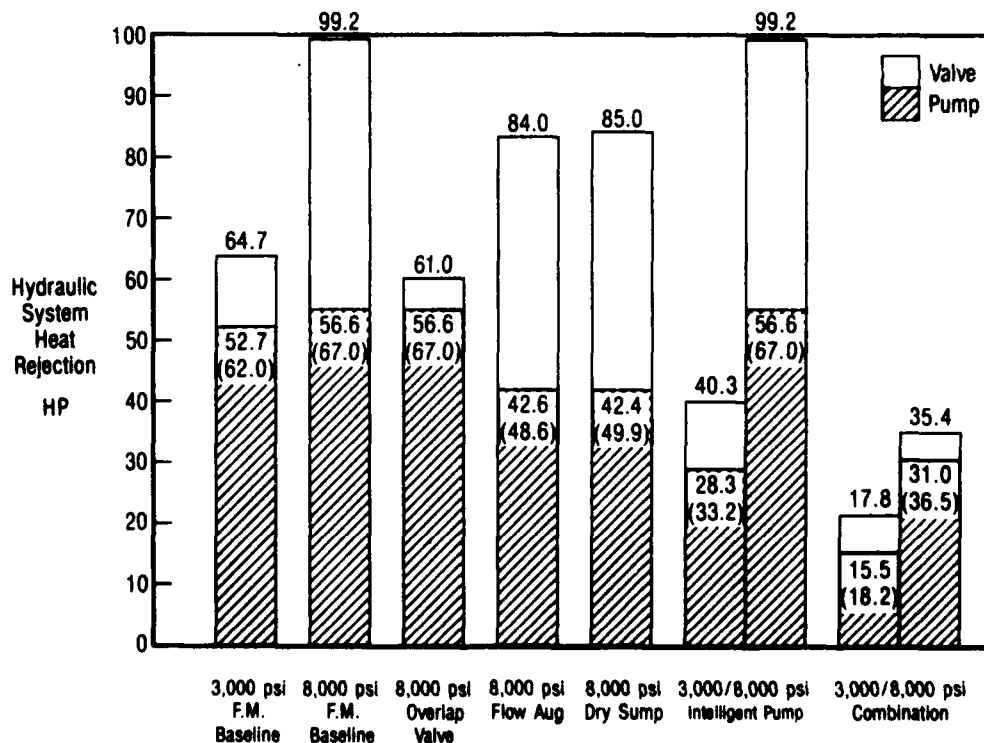
- 1) Conduct Trade Offs and Select Test Candidates
- 2) Define Laboratory Test System

3.1 CONCEPT DESCRIPTION AND EVALUATION TECHNIQUE

3.1.1 Concept Description - As mentioned, four candidate concepts were evaluated analytically in Phase II:

- o Overlap valves
- o Flow augmentation with load recovery valves
- o Dry sump pump
- o Intelligent pump

The predicted savings in heat rejection and energy for each concept are shown in Figure 101. The crosshatched section of each bar represents the total pump heat rejection of the four aircraft pumps.



Note: (xx) indicates the theoretical pump heat rejection, bar is the actual value used in thermal analysis.

Figure 101
F-15 SMTD Steady-State Hydraulic System
Heat Rejection vs. Aircraft Configuration

The pump heat rejection values in parentheses represent the total input horsepower to the pumps with zero system flow. These values were reduced by 15 percent for input into the thermal analysis computer program. Tests have shown that heat apparently is dissipated from the pump by conduction, convection and radiation before it reaches the system. The valve leakage heat input into the systems is represented by the noncrosshatched portion of the bar.

The baseline 3,000 psi and 8,000 psi configurations have nearly the same pump heat rejection, because the 8,000 psi pumps were sized smaller. However, since the valves are "line-to-line," their heat generation at 8,000 psi increased significantly.

Overlap Valves - Overlap valves (0.010 overlap), Figure 102, reduce the valve heat generation to approximately 10 percent of the line-to-line valves.

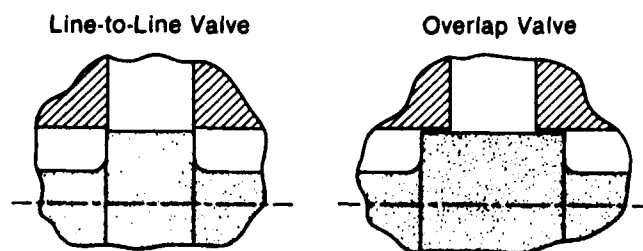


Figure 102
Overlap Valve

Flow Augmentation With Load Recovery Valves - Flow augmentation with load recovery valves, Figure 103, reduced the pump flow demand when the actuator was operating at low load/no load/assisting loads. The concept used a jet pump to recirculate return flow to the actuator pressure side. The lower flow demand on the pump allowed use of a smaller pump with smaller distribution system lines and lower heat rejection, see Figure 101.

Dry Sump Pump - The dry sump pump concept, Figure 104, reduced the windage heat generation associated with running the pump wet in a full case at return system pressures. By scavenging the oil from the case with a small positive displacement pump, the case pressure of the system pumps would be reduced to about 1 psia, and the total pump heat rejection would be reduced by 20 percent to 30 percent as shown in Figure 101. The dry sump concept does add a small amount of weight to the pump. However, this weight would be offset because about 25 cubic inches of fluid could be eliminated from the pump case. With CTFE fluid and four pumps per aircraft, this equated to 6.5 lbs.

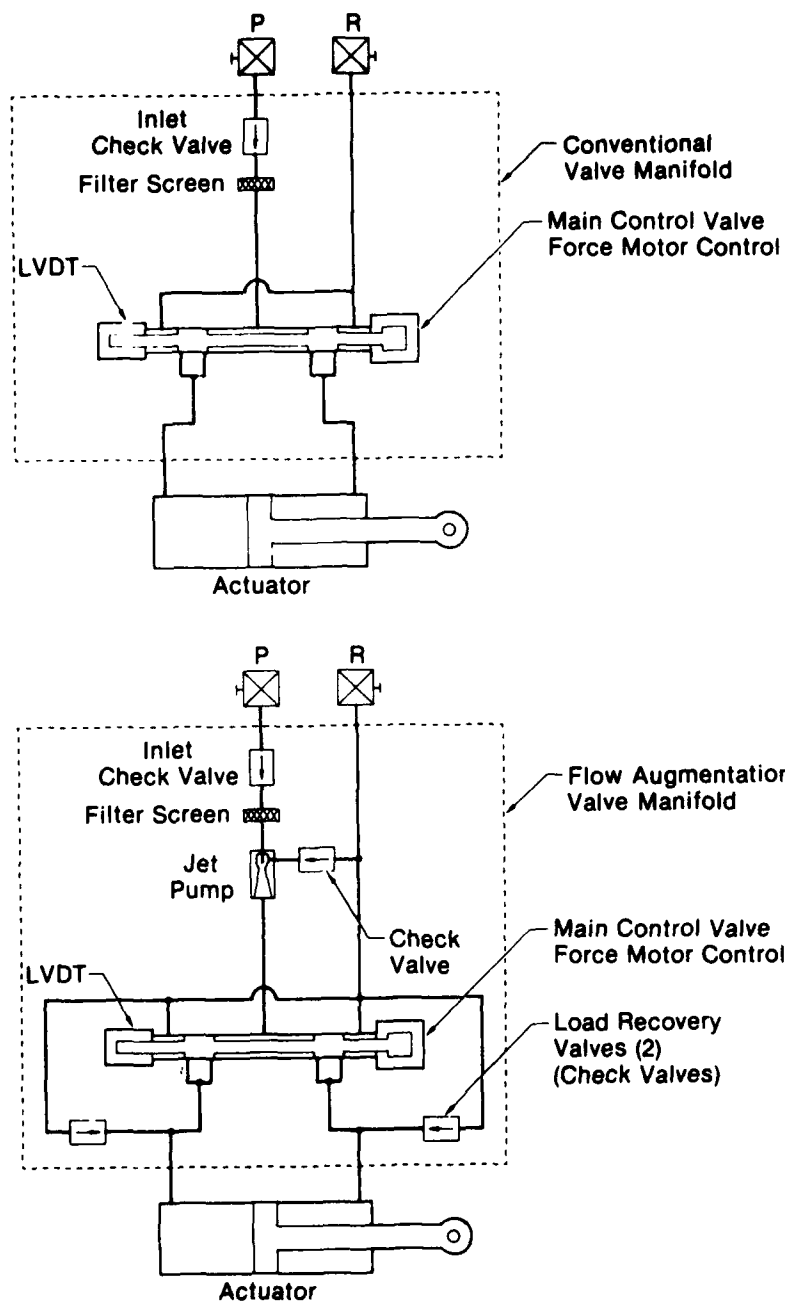


Figure 103
Flow Augmentation/Load Recovery Valve

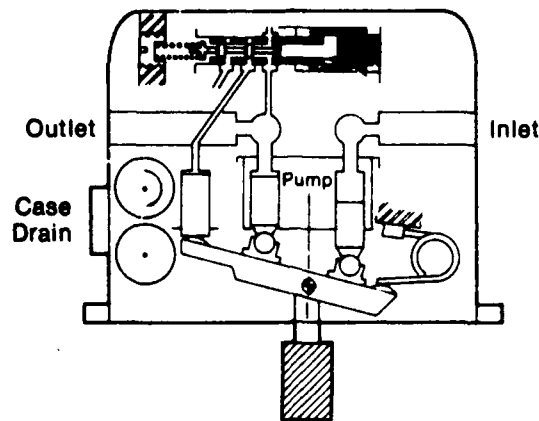


Figure 104
Dry Sump Pump

Intelligent Pump - The intelligent pump concept, Figure 105, offered variable pressure, (between 3,000 and 8,000 psi), and variable flow, because hydraulic flight control actuators and utility functions do not require full system operating pressure at all times. A computer was used to control the pump, based on valve error signals. Although pump complexity was increased, benefits included reduced heat rejection and possible increased reliability of system seals, because the pump would be at the 3,000 psi operating pressure 93 percent of the time. However, there is concern about fatigue damage due to the cycling from 3,000 to 8,000 psi, which must be considered during component design.

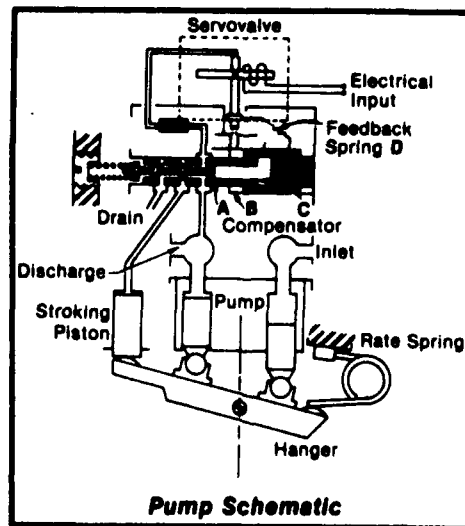


Figure 105
Intelligent Pump

Other Concepts - As mentioned, variable area actuators, variable displacement motors and pressure intensifiers were considered but were not selected for further evaluation.

The variable area actuator is based on the concept of dual, tandem configuration actuators, with extend and retract areas sized to allow pressurizing one side at a time according to load and rate demands. This concept added complexity and was considered redundant to the use of the intelligent pump.

The variable displacement motor is a servohydraulic type, to achieve maximum efficiency. This concept would increase weight and complexity, and was also redundant to the use of the intelligent pump. It was decided not to study the concept further.

The pressure intensifier approach would provide high pressure, allowing smaller area actuators and can reduce the size of the central system pump and other components, thus saving weight. This concept increases complexity with reduced reliability. Because the concept has not been developed to a point where it could be integrated with an actuator in the time frame for this program, it was not included in the trade studies.

3.1.2 Concept Evaluation Technique

a. Hydraulic System Thermal Analysis - Thermal analyses were performed on the concepts mentioned in Section 3.1.1. The ground rules and thermal design criteria are listed below:

- o Fuel temperature to engine less than 195°F
- o Fuel tank temperature less than 135°F
- o Hydraulic fluid temperature less than 275°F
- o No modifications to aircraft fuel system (use existing fuel recirculation capability)
- o Add ram air heat exchangers as necessary
- o No air circulation in engine compartment
- o No attempt made to control hydraulic fluid temperature to engine nozzle actuators
- o Eight engine nozzle actuators lumped into one model for rates and surface areas

The first task was to perform a hydraulic thermal analysis on the baseline 3,000 psi hydraulic system with MIL-H-83282 fluid. A similar hydraulic thermal analysis was then performed on the LECHT baseline 8,000 psi hydraulic system with CTFE fluid.

Subsequently, the low energy consumption concepts were evaluated individually, then in combination in the 8,000 psi CTFE baseline hydraulic system. Temperature levels were established and heat rejection and additional heat exchanger weight were determined for each concept. These weights were utilized in the life cycle cost study.

(1) Computer Model/Approach - The F-15 Fuel/Hydraulic Heat Transfer Program was used to determine fluid temperatures in the hydraulic system during a typical STOL mission flight.

Figure 106 illustrates the hydraulic system/fuel system heat exchanger interface. The transient thermal models developed for the Utility, PC-1 and PC-2 hydraulic systems, are shown in Figures 107, 108 and 109. These thermal models incorporate appropriate tubing diameters and lengths, materials, hydraulic flow rates and component heat rejection rates. The severe environment for the engine nozzle actuators was defined by the engine manufacturer, see Figure 110. The model considers convection and radiation heat transfer to the ambient environment, components and the hydraulic fluid. It also considers conduction through insulation on the engine nozzle actuators.

A typical STOL mission profile was established, see Figure 111. Temperature - time histories were calculated for the hydraulic fluid at various points in the system. These temperatures were used to determine the additional heat exchanger requirements needed to keep the hydraulic fluid and fuel temperatures below the maximum allowable levels.

It should be noted that no attempt was made to control engine nozzle actuator temperatures in this part of the analysis. Figure 112 shows the hydraulic flow demand vs. time for the chosen mission profile. The flows and leakages used in the thermal model were the average values. The peak transient flows occurred for very short time durations and were not considered in this analysis, but they did indicate the peak pump flow demand during a mission segment.

(2) Heat Exchanger/ECS Weight - After the maximum temperature levels were determined, the additional heat exchanger requirement was assessed. The production F-15 aircraft was designed such that when the hydraulic oil delivery temperature increased to 225°F, the oil was bypassed to a fuel/oil heat exchanger maintaining a fluid temperature less than 225°F. Aircraft fuel temperature levels were maintained less than 195°F. These criteria were used for the initial thermal analysis. However, the additional heat loads of the SMTD configuration for the 3,000 and 8,000 psi baselines and concepts, extended the fuel beyond its heat sink capacity.

A ram air heat exchanger was added to handle the heat load. It should be noted that the ground operation was the most severe from the hydraulic heat standpoint, and the fan and motor comprised the largest portion of the cooling system weight.

b. Hydraulic System Life Cycle Cost Analysis - LCCs were developed on a subsystem basis, (hydraulics and related equipment), and on a system basis, (total aircraft), using the procedure shown in Figure 113. The LCC of the hydraulics and related equipment was calculated using the RCA PRICE model. Development, production and support costs were estimated for the F-15.

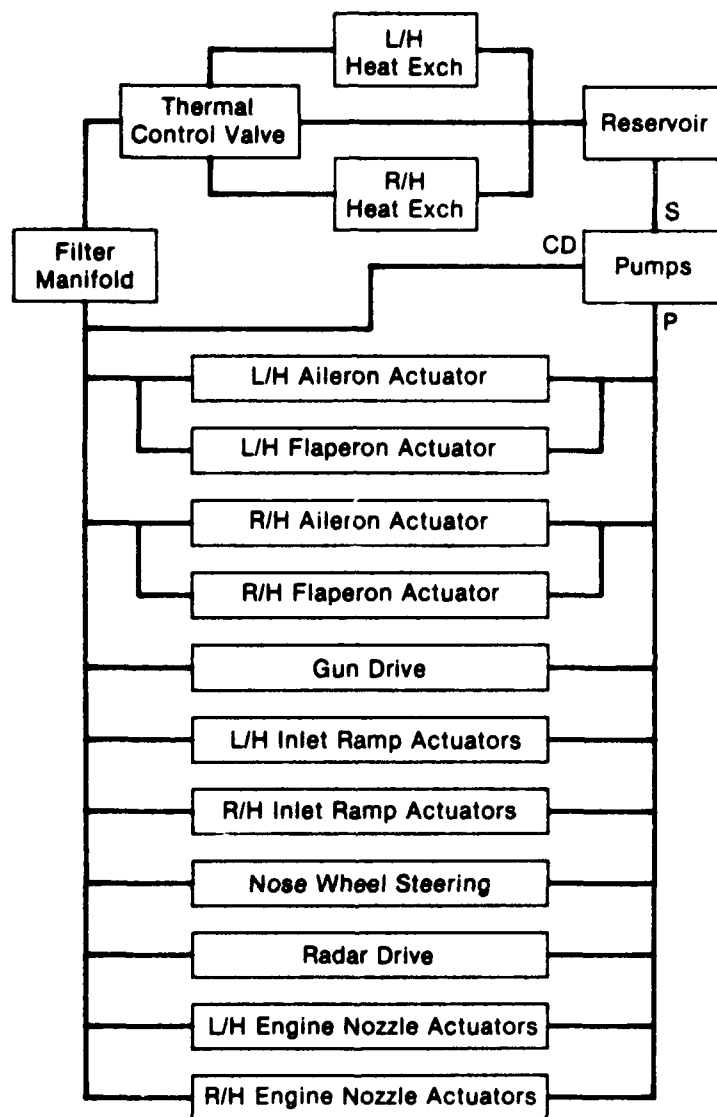


Figure 107
CTFE 8,000 psi F-15 SMTD
 Utility Hydraulic System Thermal Model

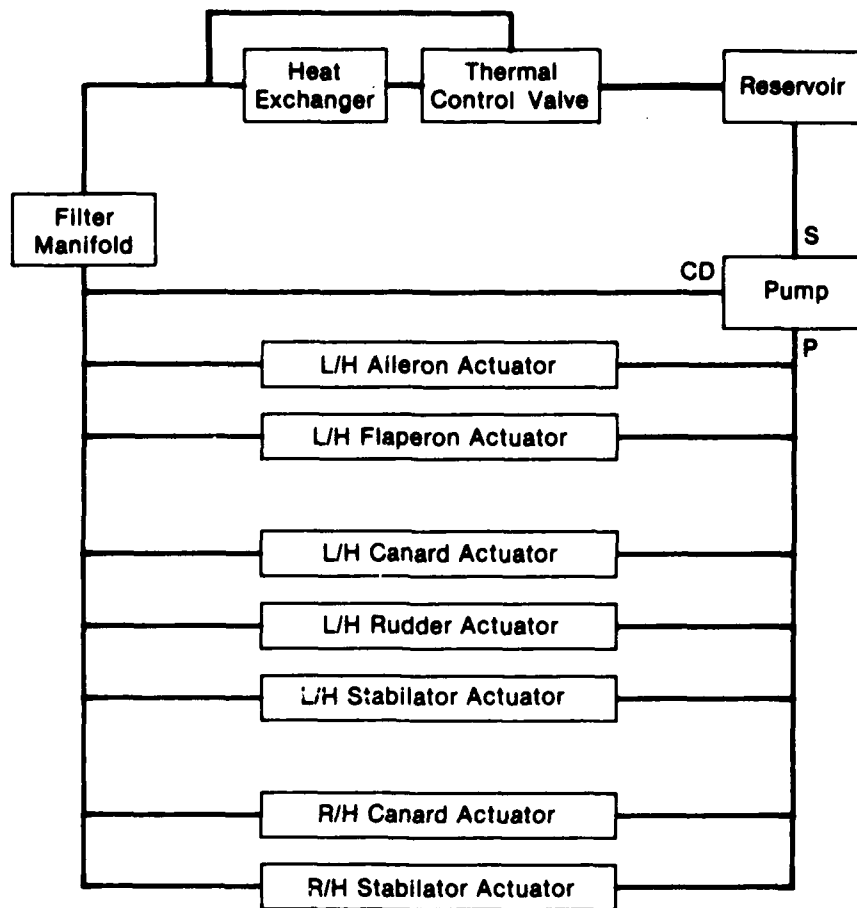


Figure 108
CTFE 8,000 psi F-15 SMTD
PC-1 Hydraulic System Thermal Model

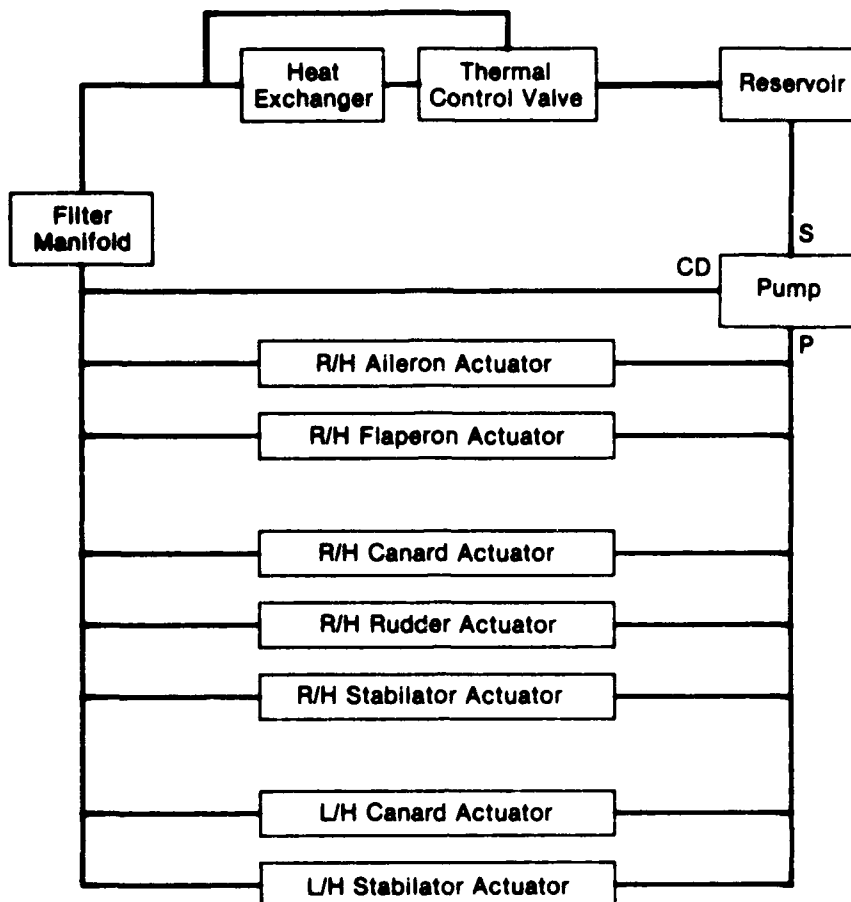


Figure 109
CTFE 8,000 psi F-15 SMTD
PC-2 Hydraulic System Thermal Model

- Engine Wall
 - 700°F Wall
 - 1/2 in. Insulation
 - Contact With 40% Actuator Surface Area
- Engine Compartment
 - 495°F Ambient Air
 - Natural Convection
 - 60% of Actuator Surface Area Exposed

Figure 110
Thermal Design Parameters
Engine Nozzle Actuators

Flight Phase	Time in Phase (sec)	Cumulative Time (sec)
Taxi	900	900
STOL	14	914
Climb and Retract Gear	399	1,313
Cruise	1,336	2,649
Combat	612	3,261
Cruise	2,557	5,818
Descent	1,072	6,890
STOL	180	7,070
Reverse	7	7,077

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Figure 111
F-15 8,000 psi SMTD Mission Profile

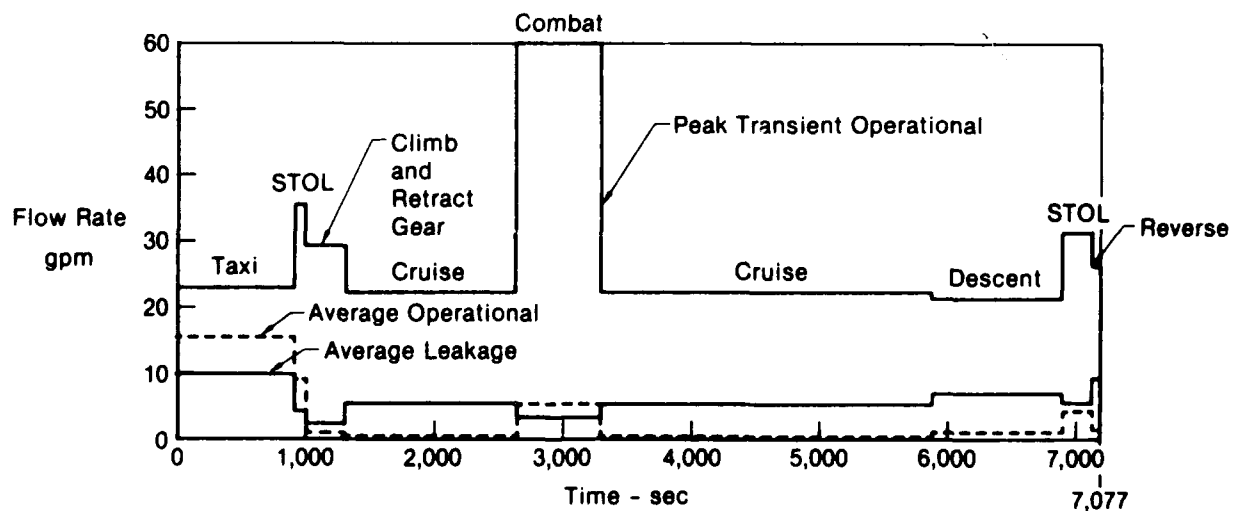


Figure 112
F-15 8,000 psi SMTD Baseline Flow Demand

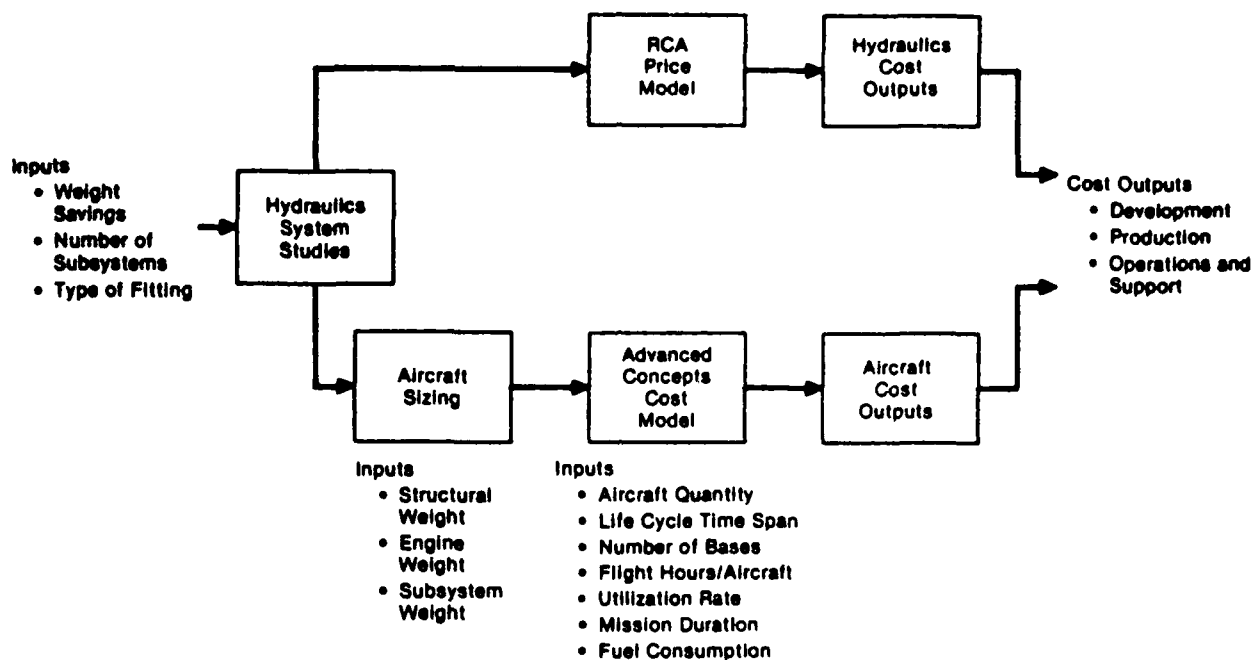


Figure 113
Cost Analysis Procedure

The LCC was calculated using a method that resized the aircraft to maintain constant performance for changes in hydraulic system weight. The F-15 SMTD aircraft with its 3,000 psi hydraulic subsystem was used to establish the baseline.

(1) Reliability and Maintainability - F-15 Reliability and Maintainability data was used from the USAF 66-1 maintenance data collection system. For the 3,000 and 8,000 psi hydraulic systems, reliability was considered to be the same. However, hydraulic component cost was increased to achieve this. Figure 114 shows the reliability and maintainability assessment techniques and parameters used to develop data. Figure 115 shows the MFHBF and MTTR for the 8,000 psi baseline. Figure 116 shows the components that were affected when the low energy consumption concepts were analyzed. These numbers were derived through component vendor inputs.

- Techniques Utilized in F-15, F/A-18, AV-8B and Future Programs
- Like/Similar Evaluation of Field Data
 - AFM 66-1, Navy 3M, MCAIR Component Life Evaluation and Reliability (CLEAR)
 - Vendor Predictions
 - Logistics Support Analysis (LSA) Predictions Per MIL-STD-1388
- Reliability Develops MTBF
- Maintainability Develops MTBMA, MTTR, MMH/FH
 - All Facets of Integrated Logistics Support (ILS) Evaluated
 - Concepts Verified
 - Optimum Repair Level Analysis (ORLA), Level of Repair Analysis (LORA)
 - Modeling Logic Control Output Module (LCOM), Comprehensive Aircraft Support Effectiveness Evaluation (CASEE), Etc.)
- Data Input to Operating and Support (OPS) Analysis
 - Life Cycle Cost Derived

Figure 114
Reliability and Maintainability Assessment

Actuators				
Application	Type*	Quantity/ Aircraft	Reliability MFHBF	Maintainability MTTR
Flight Control				
Rudders	SWFM	2	1,844	6,187
Ailerons	SWFM	2	900	4,350
Flaperons	SWFM	2	900	4,350
Canards	SWFM	2	227	6,235
Stabilators	SWFM	2	227	6,235
Engine Nozzle				
Upper Vane	SWFM	2	838	5,000
Lower Vane	SWFM	2	838	5,000
Convergent Flap	SWFM	4	419	5,000
Outboard Flap	SWFM	4	419	5,000
Inboard Flap	SWFM	4	419	5,000
Utility				
Canopy Lock	Sim	1	7,915	5,953
Refuel	Sim	1	79,151	3,160
Arresting Hook	Sim	1	8,795	3,427
ECS	Sim	1	79,151	4,960
NLG Uplock	Sim	1	3,958	3,783
MLG Uplock	Sim	2	79,151	3,907
Canopy Main	Sim	1	1,365	3,408
Air Induction Bypass Door	SWFM	2	5,863	5,444
NLG Retract	Sim	1	5,654	5,000
MLG Retract	Sim	2	4,947	5,000
Air Induction First Ramp	SWFM	2	4,947	6,004
Air Induction Diffuser	SWFM	2	13,192	6,576
Speedbrake	Sim	1	5,654	3,196
Flight Control Switching				
Ailerons	Mech	2	2,553	5,893
Canards	Mech	2	4,059	5,893
Stabilators	Mech	2	4,059	5,000
Engine Nozzle				
Actuator Control	Mech	2	4,059	5,000

Actuators				
Application	Type*	Quantity/ Aircraft	Reliability MFHBF	Maintainability MTTR
Utility				
Canopy	Sol	1	1,885	5,451
Refuel	Sol	1	19,788	3,520
Arresting Hook	Sol	1	11,307	6,000
ECS	Sol	1	39,576	6,250
NLG				
Check Valve	Sim	1	19,788	6,250
Linear Directional Control	Sol	1	79,151	7,700
Restrictor	Sim	2	158,302	5,000
MLG				
Restrictor	Sim	2	158,302	5,000
Linear Directional Control	Sol	1	11,307	6,828
Brake	Mech	1	13,192	2,708
Emergency Extend	Sol	1	79,151	3,000
Miscellaneous				
Emergency Generator	Sol	1	79,151	5,000
Gun System	Sol	1	79,151	6,367
Utility				
Temperature Regulator	Mech	1	3,827	5,348
Primary HX	Mech	1	2,827	5,000
Jet Fuel Starter	Sol	1	79,151	5,000
PC-1, PC-2				
Temperature Regulator	Mech	2	1,414	3,204
Utility				
Pump	—	2	8,795	5,000
Reservoir	—	1	1,439	5,000
Primary HX	—	3	8,333	5,000
HX Fan	—	3	2,000	5,000
Canopy				
Accumulator	—	1	11,307	3,649
Jet Fuel Starter				
Hand Pump	—	1	3,598	4,170
Accumulator	—	1	682	5,149
PC-1, PC-2				
Pump	—	2	8,795	5,392
Reservoir	—	2	3,958	5,000

*Legend
SWFM = Servo With Force Motor MECH = Mechanical
Sol = Solenoid Sim = Simple

Figure 115
Reliability and Maintainability
MFHBF and MTTR 8,000 psi Baseline System Data

Concept	Components Affected	Type	Quantity Aircraft	Reliability MFHBF	Maintainability MTTR
Dry Sump Pump	Pump	—	4	8,041	5.90
Flow Augmentation	Canard Actuator	SWFM	2	600	6.54
	Stabilator Actuator	SWFM	2	600	6.54
	Aileron Actuator	SWFM	2	860	4.55
	Flaperon Actuator	SWFM	2	860	4.55
	Rudder Actuator	SWFM	2	1,657	6.88
	Pump	—	4	11,250	—
Intelligent Pump	Pump	—	4	7,820	6.06
Overlap Valve	No Change				
Combination	Canard Actuator	SWFM	2	600	6.54
	Stabilator Actuator	SWFM	2	600	6.54
	Aileron Actuator	SWFM	2	860	4.55
	Flaperon Actuator	SWFM	2	860	4.55
	Rudder Actuator	SWFM	2	1,657	6.88
	Pump	—	4	9,705	4.88

Note: SWFM - Simplex With Force Motor

Figure 116
Reliability and Maintainability Changes

(2) Ground Rules and Assumptions - The LCCs are the costs associated with full scale development, production, and operation and support in the user environment. The ground rules are as follows:

- o Constant 1985 dollars
- o 20 RDT&E aircraft
- o 500 shipsets of hardware costed
- o Support equipment not costed
- o 15 year operational life
- o 300 flying hours per aircraft per year
- o Operational deployment in three theatres
- o Seven base-intermediate maintenance locations

(3) RCA PRICE Model Analysis - The hydraulic subsystem weight categories are shown in Figure 117. The miscellaneous components included all valves, pumps, reservoirs and related attaching structures. The distribution system is comprised of tubing, fittings, clamps, brackets, etc. The ram air heat exchanger included the ram air heat exchanger, increased capacity fuel/oil heat exchanger, the fan and motor for ground cooling and installation hardware.

- Flight Control Actuators
- Engine Nozzle Actuators
- Utility Actuators
- Heat Exchangers
- Miscellaneous Components
- Distribution System
- Fluid

Figure 117
Hydraulic Subsystem Weight Categories

The cost analysis for each equipment level was conducted at an individual component level; i.e., aileron actuator, rudder actuator, switching valve, utility pump, and utility reservoir, and accumulated to the level shown in Figure 118. Each component was described in terms of independent model inputs.

- Development
- Procurement
 - Equipment
 - Initial Spares
 - Other
- Support
 - Replacement Spares
 - Maintenance Manpower
 - Other
- Total Life Cycle Cost
- Aircraft Fuel Costs
- Total Life Cycle Cost

Note: (1) Fuel savings are based on total hydraulic and related weight savings

Figure 118
Hydraulics and Related Equipment Cost
Millions of FY 85 Dollars

The unit production hydraulic subsystem costs vary with the aircraft configuration. The costs are categorized as shown in Figure 117. Cost values were determined from independent model inputs which included:

- o Precision values - governing tolerances for the fabricated parts
- o Number of parts - quantity of fabricated parts contained in a component
- o Distance - length over which precision must be maintained
- o Materials - Cost per unit of material from which component was fabricated

The sum total of all the component procurement costs yielded the unit flyaway cost. The unit flyaway costs for each concept is shown in Section 3.3.

(4) ACCM Analysis - After the weight and cost impact was determined for the hydraulic system, growth factor analysis was used to determine the total aircraft weight. This weight was distributed to structure, fuel, engine and subsystems. The resulting weight breakdowns were derived for the 3,000 psi MIL-H-83282 and CTFE fluid baselines, and the 8,000 psi CTFE baseline. Each low energy consumption concept was analyzed in a similar manner. The data is presented in Section 3.3.1.

3.2 BASELINE HYDRAULIC SYSTEMS

Phase I established baseline hydraulic systems with MIL-H-83282 and CTFE fluid at 3,000 psi, then established a baseline system with CTFE fluid at 8,000 psi.

The baseline parameters were:

- o Total hydraulic system weight
- o Heat exchanger weight
- o Reliability
- o Maintainability
- o System complexity
- o Life cycle cost
 - Hydraulic subsystem level
 - Total aircraft system level

Efforts performed in Phase II reevaluated the baseline criteria. Modifications were made because of discrepancies, so thermal analyses and life cycle cost analyses were rerun.

3.2.1 Baseline Thermal Modifications - Changes made to the baseline thermal models were:

- o Updated the specific heat (Cp) to new Monsanto values
- o Added the heat from case drain flow
- o Changed the pump heat distribution between pump outlet and case drain
- o Changed the heat exchanger design condition from combat operation to ground idle taxi

Updated CTFE fluid properties were supplied to MCAIR at the beginning of Phase II. They showed that at high temperatures, the specific heat was about one half of that used in Phase I.

The case drain heat was not included in the total pump heat rejection analysis. The computer model did not account for this parameter.

Also, the distribution of total pump heat was corrected. The proper ratio for the baseline study is two-thirds to the case drain and one-third to the pump discharge.

The ram air heat exchanger design criteria was reevaluated. The decision was made to maintain a constant design condition for all the concepts. This particular condition may not be optimum for each concept, but served as a uniform means of comparison.

3.2.2 Life Cycle Cost Modifications - The following LCC discrepancies were found in the Phase I analysis and corrections were made in the Phase II LCC analysis:

- o Total aircraft weight accounted for hydraulic system weight twice
- o Fluid cost incorrect for CTFE (used MIL-H-83282 cost)
- o Aluminum procurement costs were used instead of titanium
- o Increased manufacturing complexity for 8,000 psi hardware (tighter tolerances/clearances)
- o Phase I added 0.73 MTTR for documentation - this was already included in the maintainability data

3.3 TASK 1 - CONDUCT TRADEOFFS

3.3.1 Baseline Systems - Hydraulic system operating pressures of 3,000 and 8,000 psi were studied using both MIL-H-83282 and CTFE fluid to establish baselines. These configurations were analyzed three ways: (1) thermal analysis, (2) weight analysis, and (3) life cycle cost analysis.

a. Thermal Analysis - The thermal analysis for the baseline concepts was divided into two segments. One was the determination of the pump heat rejection and the other was determination of the system average operational and the system average leakage flow rates.

The system pump heat rejection is shown in Figure 101 for the 3,000 psi baseline MIL-H-83282 and CTFE fluid systems. As shown, this heat equates to 62.0 horsepower per aircraft or 15.50 horsepower per pump, and was established in Phase I. The 8,000 psi baseline pump heat rejection was 67.0 horsepower per aircraft or 16.64 horsepower per pump.

The case drain flow for each baseline hydraulic pump was assumed to be 2 gpm. The pump heat was divided into two parts: (1) the heat rejection in the case drain flow, and (2) the heat rejection in the pump discharge. For the baseline configurations, the distribution was assumed to be two-thirds/one-third relationship, as shown in Figure 119.

Configuration	Pump Heat Rejection	H.R. Accounted for (%)	Case Drain Flow	H.R. to Case Drain (%)	H.R. to Discharge (%)	Actuator Operational Flows	Actuator Leakage Flows	System Pressure
8,000 psi Baseline	16.64 hp	14.14 hp (85%)	2.0 gal./min	9.47 hp (67%)	4.67 hp (33%)	Shown on Table	Shown on Table	8,000 psi
0.010 Overlap Valve	16.64 hp	14.14 hp (85%)	2.0 gal./min	13.15 hp (93%)	1.0 hp (7%)	Same as Baseline	Decreased. Shown on Table	8,000 psi
Flow Augmentation	12.14 hp	10.32 hp (85%)	1.5 gal./min	6.91 hp (67%)	3.41 hp (33%)	Same as Baseline	Same as Baseline	8,000 psi
Dry Sump Pump	12.48 hp	10.60 hp (85%)	2.0 gal./min	7.10 hp (67%)	3.50 hp (33%)	Same as Baseline	Same as Baseline	8,000 psi
Intelligent Pump	16.64/8.32 hp (8,000/3,000)	14.14/7.07 hp (85%)	2.0 gal./min, 1.0 gal./min (8,000/3,000)	9.47/4.74 hp (67%)	4.66/2.33 hp (33%)	Shown on Table	Shown on Table	8,000/3,000
Combinations	9.11/4.56 hp (8,000/3,000)	7.74/3.87 hp (85%)	1 gal./min, 0.7 gal./min (8,000/3,000)	5.16/2.59 hp (67%)	2.55/1.28 hp (33%)	Shown on Table	Shown on Table	8,000/3,000

Figure 119
8,000 psi Baseline Thermal Model Parameters

Based on pump heat rejection tests performed at MCAIR, it was concluded that only 85 percent of the pump heat rejection could be quantifiably measured. The remaining 15 percent was assumed to be lost due to free convection and radiation. Therefore, as shown in Figure 119, the pump heat rejection at 8,000 psi was reduced to 14.14 horsepower per pump. This number was used in the thermal analysis for the baseline configuration at 8,000 psi.

The actuator operational and leakage flow rates at 3,000 psi are shown in Figures 120 and 121. This data was developed from the F-15 configuration. A 0.001-inch overlap was assumed on the valve spool, which decreased null leakage. The operational flow was determined through a study based upon the F-15 aircraft flow requirements.

Per Actuator	Actuator Flow (Unloaded)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	2.89	0.088 ¹	0.091	0.091	0.06	0.439	0.06	0.06	0.202	0.030
Flaperon	3.48	0.35 ¹	—	0.363	—	0.132	—	—	0.326	0.037
Rudder	3.64	0.053 ¹	0.114	0.038	0.038	0.5523	0.038	0.038	0.380	0.380
Stabilator	30.92	0.70 ¹	2.91	1.612	0.645	4.69	0.645	1.29	2.91	0.322
Canard	26.15	0.70 ¹	2.046	0.818	0.272	3.97	0.272	0.818	1.364	0.272
Utility System										
Gun	29.0	—	—	—	—	0.284	—	—	—	—
Steering	2.41	1.61	1.71	—	—	—	—	—	—	1.71
Radar	1.70	1.70	1.70	1.70	1.70	1.70	1.70	1.70	1.70	1.70
Nozzles	—	—	16.0	0.544	0.162	0.431	0.162	0.103	2.88	1.44
Ramps	105.52	—	0.097	0.136	0.136	3.79	0.136	0.136	0.97	—
Aileron	5.78	0.010 ¹	0.180	0.180	0.121	0.977	0.121	0.121	0.404	0.06
Flaperon	6.96	0.042 ¹	—	0.726	—	0.264	—	—	0.653	0.073

¹ = leakage

Note: Flows are mean values over flight phase in GPM

Figure 120
F-15 SMTD
Operational Flow 3,000 psi Baseline

Per Actuator	Null Leakage (per Aircraft)	Flight Phase								
		Taxi	STOL	CHmb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	0.088	0.088	0.061	0.061	0.07	—	0.07	0.07	0.03	0.079
Flaperon	0.35	0.35	0.35	—	0.35	0.016	0.35	0.35	0.035	0.315
Rudder	0.053	0.053	0.037	0.047	0.047	—	0.047	0.047	—	—
Stabilator	0.70	0.70	0.07	0.35	0.56	0.14	0.56	0.42	0.07	0.63
Canard	0.70	0.70	0.175	0.49	0.63	0.56	0.63	0.49	0.35	0.63
Utility System										
Gun	0.61	0.61	0.61	0.61	0.61	—	0.61	0.61	0.61	0.61
Steering	0.08	0.02	0.02	—	—	—	—	—	—	0.02
Radar	—	—	—	—	—	—	—	—	—	—
Nozzles	2.80	2.80	—	—	—	—	—	1.40	2.54	2.80
Ramps	1.05	1.05	1.043	1.040	1.040	0.858	1.040	1.040	1.043	1.05
Aileron	0.175	0.175	0.123	0.123	0.14	—	0.14	0.14	0.058	0.158
Flaperon	0.70	0.70	0.70	—	0.70	0.70	0.70	0.70	0.70	0.70

Note: Flows are mean values over flight phase in GPM

Figure 121
F-15 SMTD
 Leakage Flow 3,000 psi Baseline

The data for the 8,000 psi operational and leakage flow is shown in Figures 122 and 123 respectively. This data was generated in a similar fashion as the 3,000 psi flow data.

Per Actuator	Actuator Flow (Unloaded)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	1.08	0.228 ¹	0.034	0.034	0.023	0.164	0.023	0.023	0.076	0.011
Flaperon	1.31	0.875 ¹	—	0.137	—	0.051	—	—	0.122	0.014
Rudder	1.37	0.175 ¹	0.043	0.015	0.015	0.208	0.015	0.015	0.143	0.143
Stabilator	11.60	1.75 ¹	1.092	0.605	0.242	1.732	0.242	0.484	1.092	0.121
Canard	9.81	1.75 ¹	0.768	0.307	0.102	1.489	0.102	0.307	0.512	0.102
Utility System										
Gun	10.88	—	—	—	—	0.107	—	—	—	—
Steering	0.90	0.64	0.64	—	—	—	—	—	—	0.64
Radar	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64
Nozzles	—	7.28	5.81	0.204	0.061	0.162	0.061	0.039	1.08	0.54
Ramps	5.91	—	0.006	0.008	0.008	0.213	0.008	0.006	0.006	—
Aileron	2.17	0.456 ¹	0.069	0.069	0.046	0.329	0.046	0.046	0.152	0.022
Flaperon	2.61	1.75 ¹	—	0.274	—	0.099	—	—	0.244	0.028

¹ = leakage

Note: Flows are mean values over flight phase in GPM

Figure 122
F-15 SMTD
 Operational Flow 8,000 psi Baseline

Per Actuator	Null Leakage (per Aircraft)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	0.130	0.130	0.092	0.092	0.104	—	0.104	0.104	0.042	0.118
Flaperon	0.50	0.50	0.50	—	0.50	0.50	0.50	0.50	—	0.45
Rudder	0.10	0.10	0.10	0.07	0.09	—	0.09	0.09	—	—
Stabilator	1.00	1.00	0.10	0.50	0.80	0.20	0.80	0.60	0.10	0.90
Canard	1.00	1.00	0.25	0.70	0.90	0.80	0.90	0.70	0.50	0.90
Utility System										
Gun	0.34	0.34	0.34	0.34	0.34	—	0.34	0.34	0.34	0.34
Steering	0.11	0.032	0.032	—	—	—	—	—	—	0.032
Radar	—	—	—	—	—	—	—	—	—	—
Nozzles	4.16	4.16	—	—	—	—	—	2.08	3.786	4.16
Ramps	1.56	1.56	1.55	1.528	1.528	1.326	1.528	1.55	1.55	1.440
Aileron	0.26	0.26	0.182	0.208	—	—	—	0.208	0.086	0.234
Flaperon	1.00	1.00	1.00	—	1.00	1.00	1.00	1.00	—	0.90

Note: Flows are mean values over flight phase in GPM

Figure 123
F-15 SMTD
 Leakage Flow 8,000 psi Baseline

Figures 124 through 135 show the temperatures that were computed using the thermal analyses computer model, as mentioned in Section 3.1.2.a. The graphs are arranged so the first plots show the component or actuator, etc., without any additional ram air heat exchanger. The next corresponding plot shows the temperature/time history with a ram air heat exchanger maintaining 275°F maximum hydraulic oil temperatures throughout the mission. Figures 136 through 147 show the 8,000 psi temperature time histories in a similar fashion. Figures 129, 135, 141 and 147 show the PC and utility ram air heat exchanger requirements. As shown, the temperature deltas are the amount of heat that the ram air heat exchanger must dissipate. The temperature delta was approximately 60° for most of the flight.

Shown beneath the ram air heat exchanger requirement, is the fuel temperature as it circulates throughout the aircraft during its mission. As indicated, the fuel temperature during the combat phase is significantly lower than it is throughout the rest of the mission. This is due to the high power setting of the aircraft engines and the high fuel flow requirements.

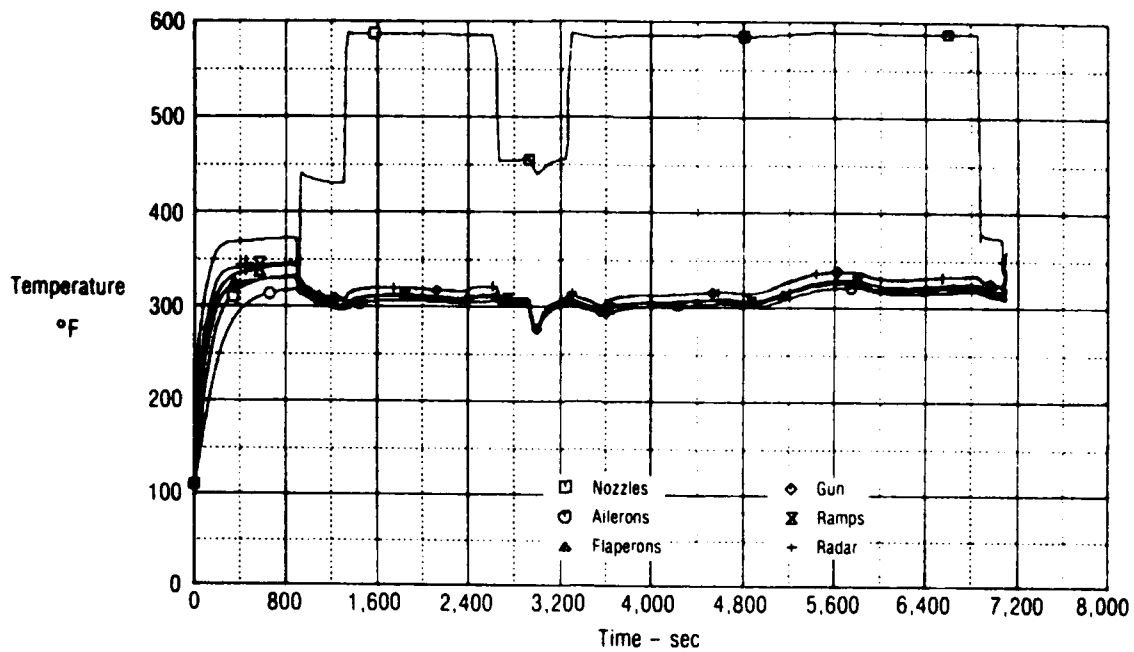


Figure 124
3,000 psi Baseline Without Ram Air HX
 Utility Actuator Exit Temperatures

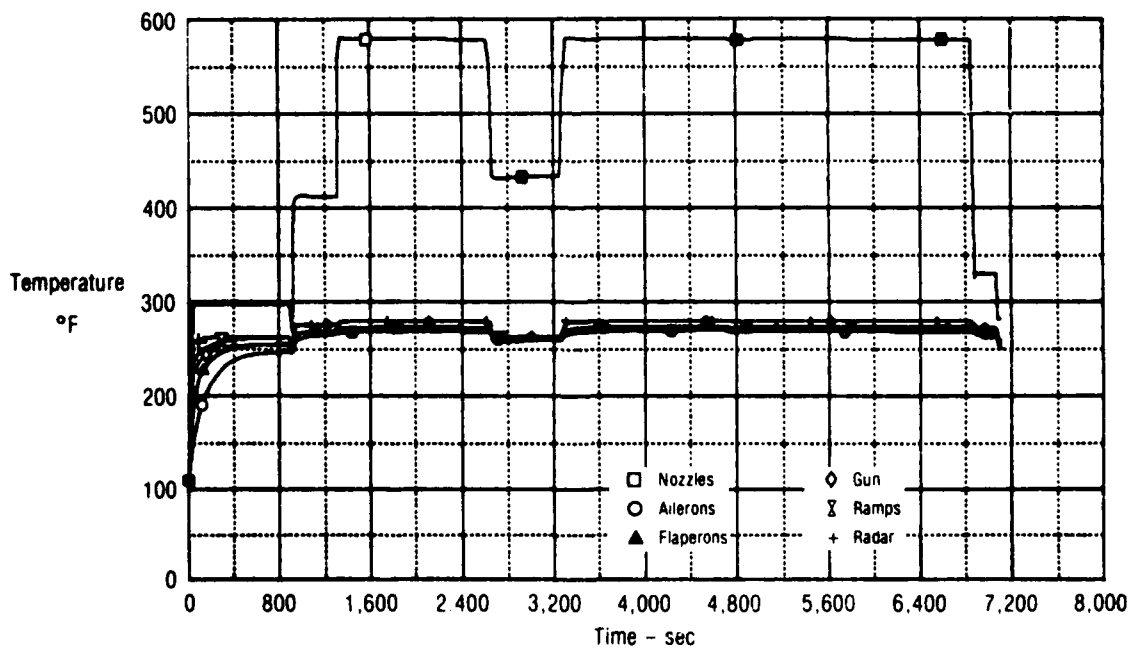


Figure 125
3,000 psi Baseline
 Utility Actuator Exit Temperatures

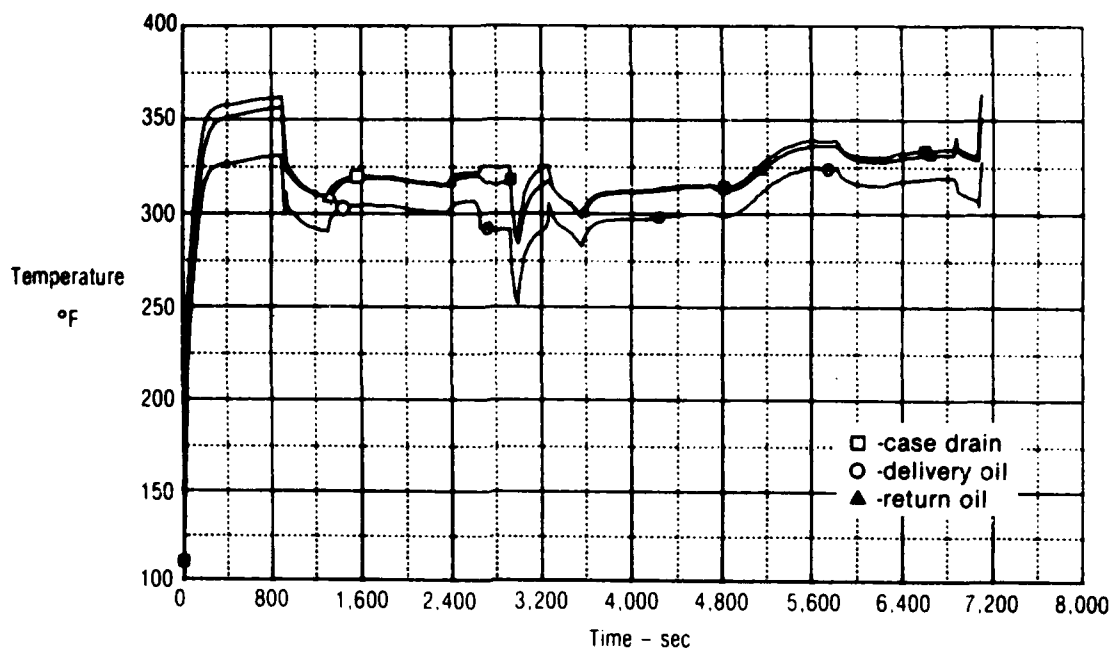


Figure 126
3,000 psi Baseline Without Ram Air HX
 Utility Pump and Return Temperatures

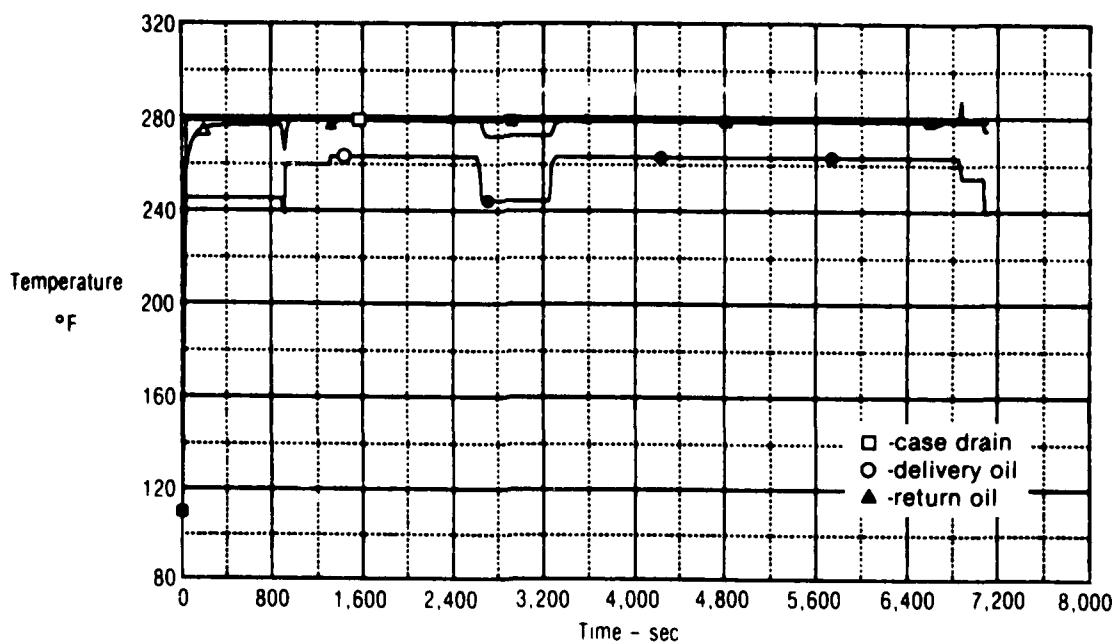


Figure 127
3,000 psi Baseline
 Utility Pump and Return Temperatures

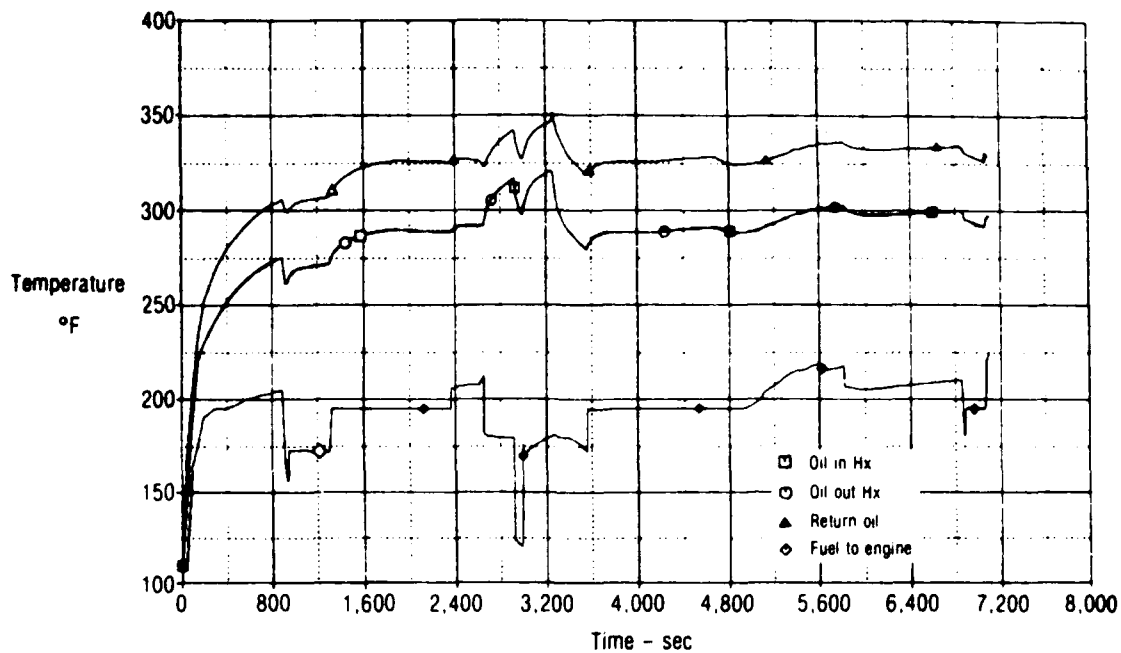


Figure 128
3,000 psi Baseline Without Ram Air HX
 Utility Ram Air HX Requirements

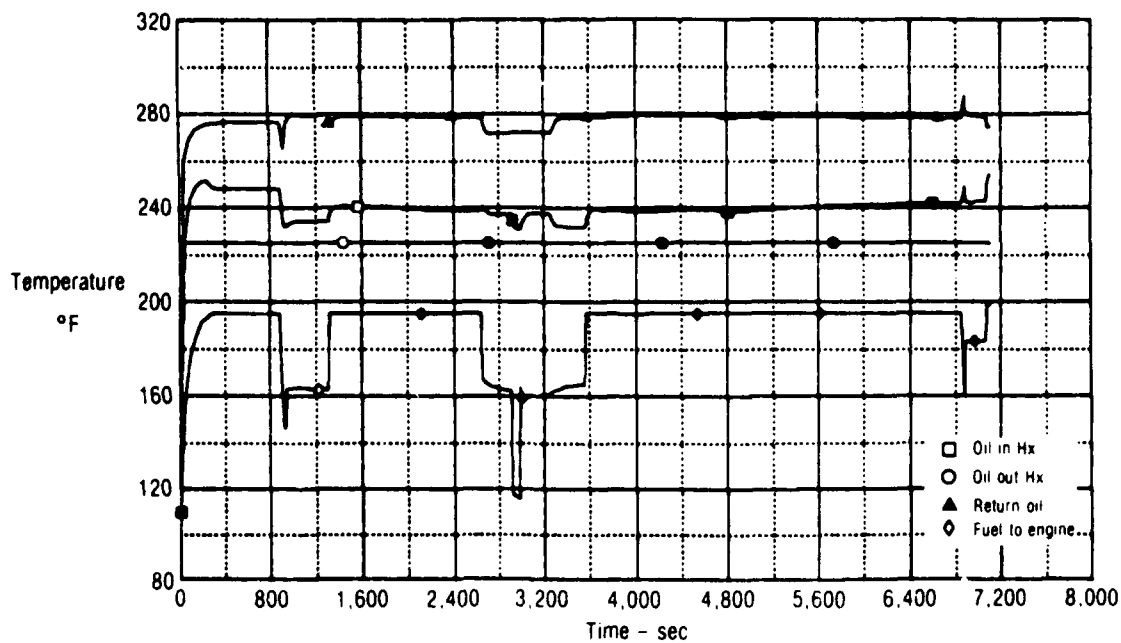


Figure 129
3,000 psi Baseline
 Utility Ram Air HX Requirements

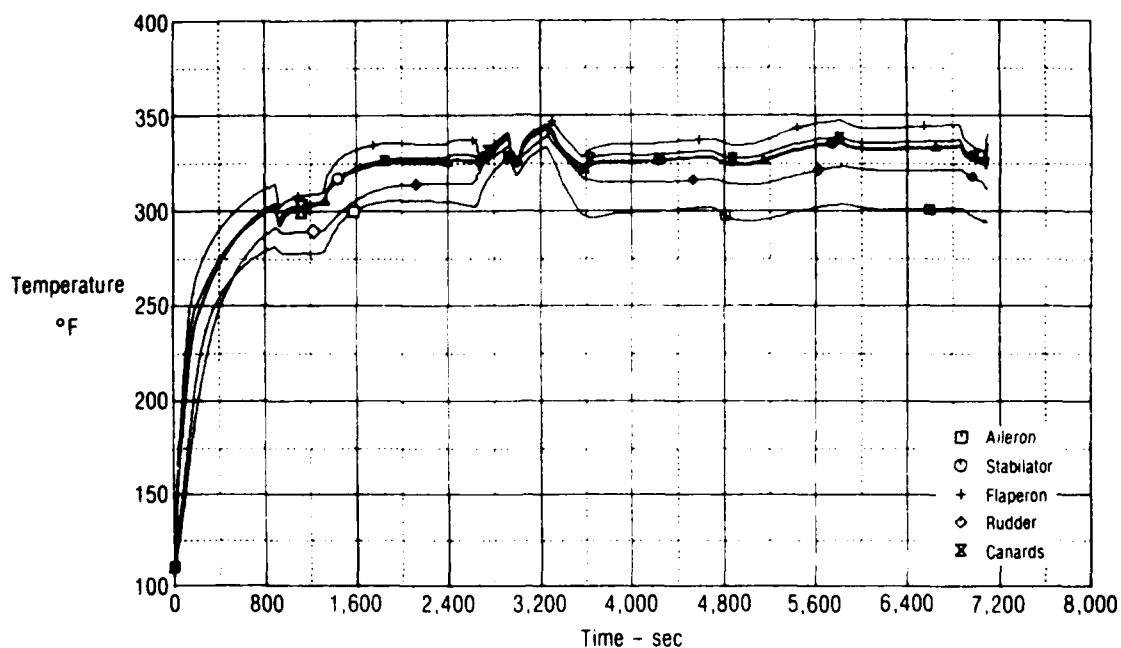


Figure 130
3,000 psi Baseline Without Ram Air HX
PC-1 Actuator Exit Temperatures

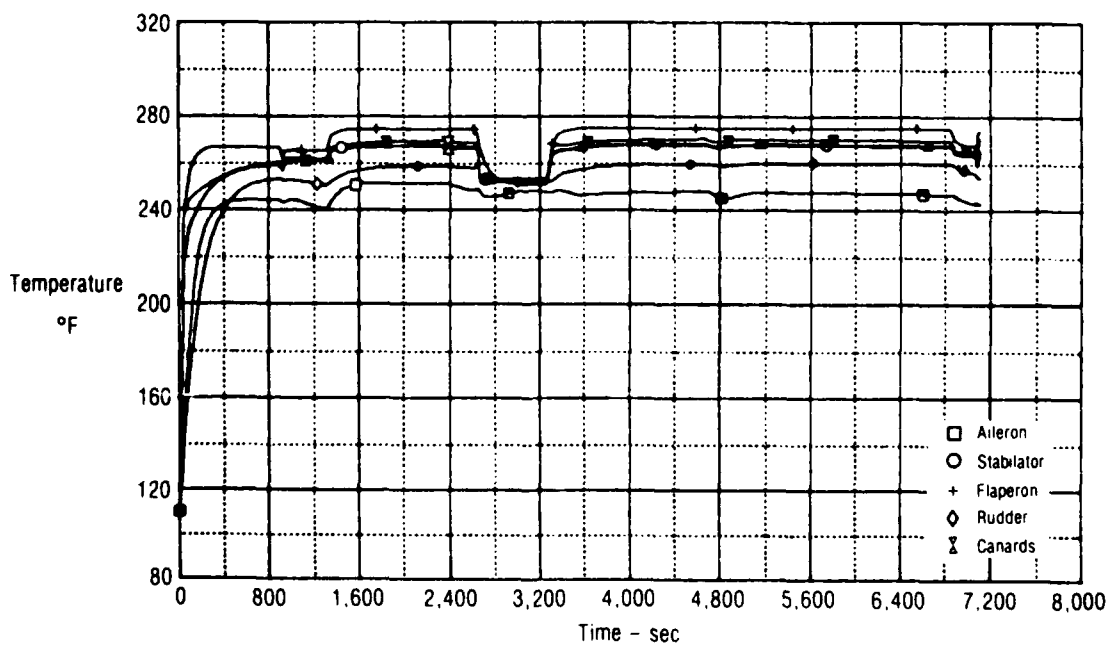


Figure 131
3,000 psi Baseline
PC-1 Actuator Exit Temperatures

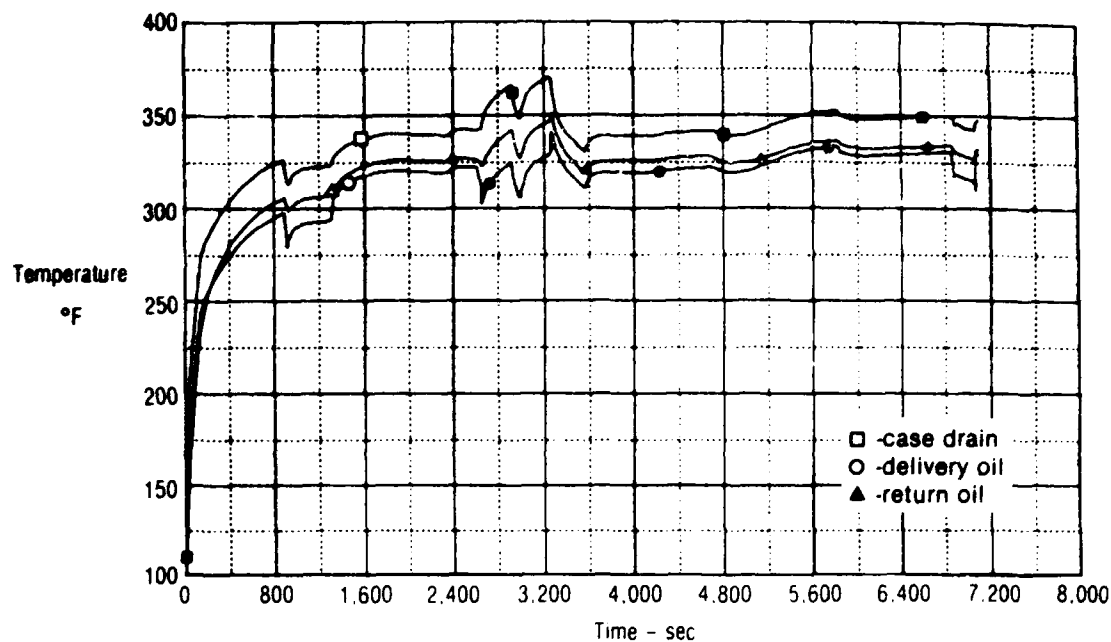


Figure 132
3,000 psi Baseline Without Ram Air HX
 PC-1 Pump and Return Temperatures

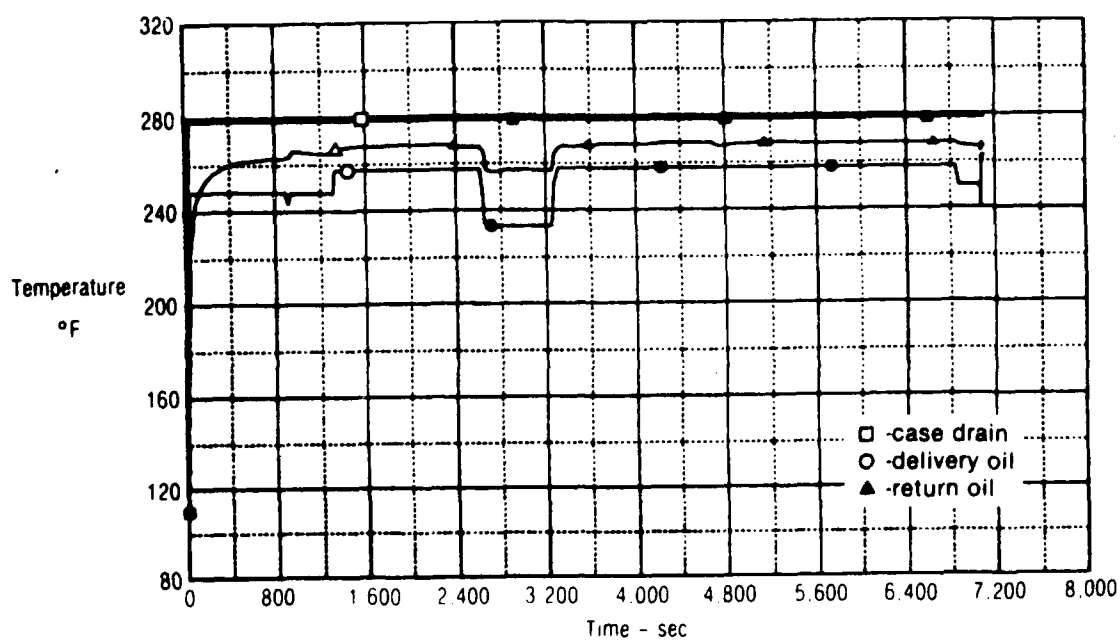


Figure 133
3,000 psi Baseline
 PC-1 Pump and Return Temperatures

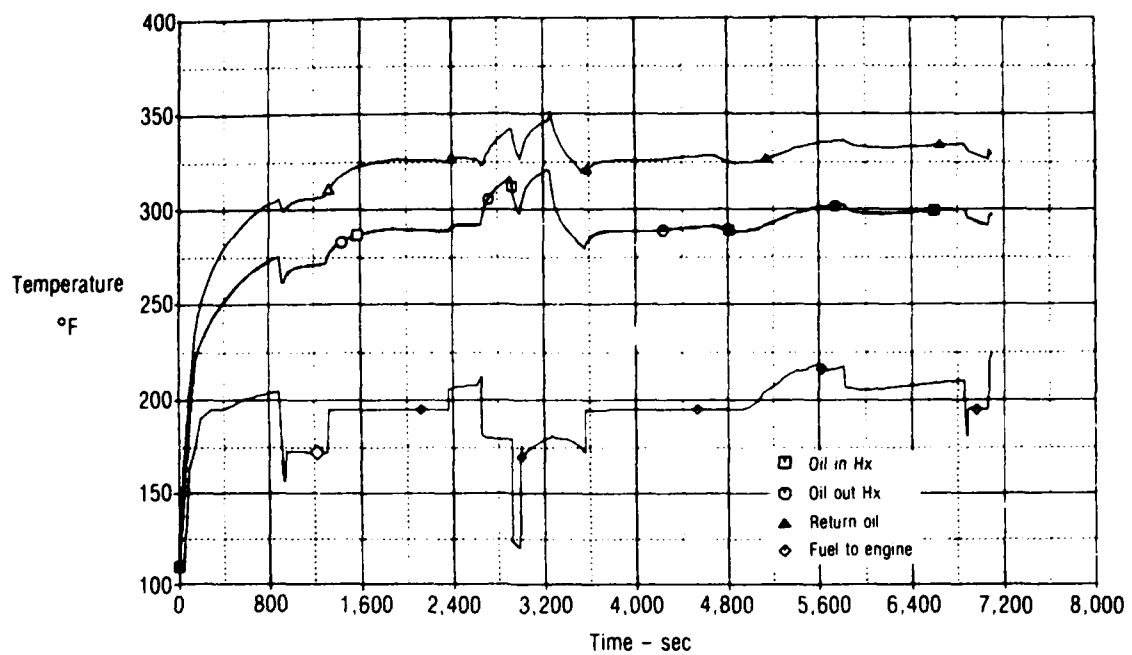


Figure 134
3,000 psi Baseline Without Ram Air HX
PC-1 Ram Air HX Requirements

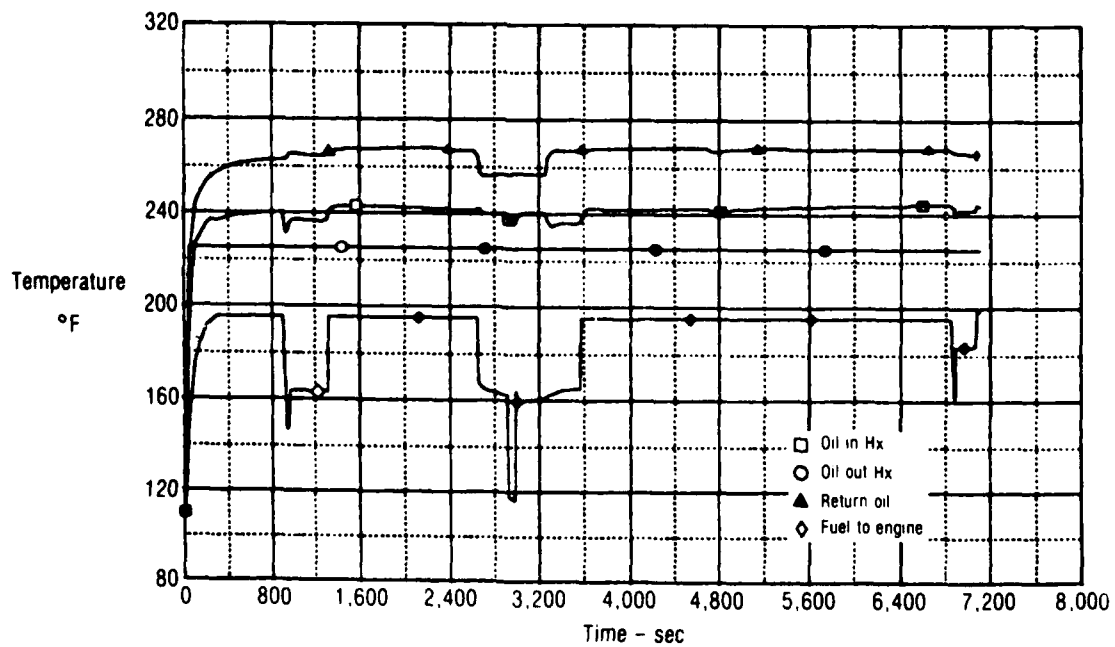


Figure 135
3,000 psi Baseline
PC-1 Ram Air HX Requirements

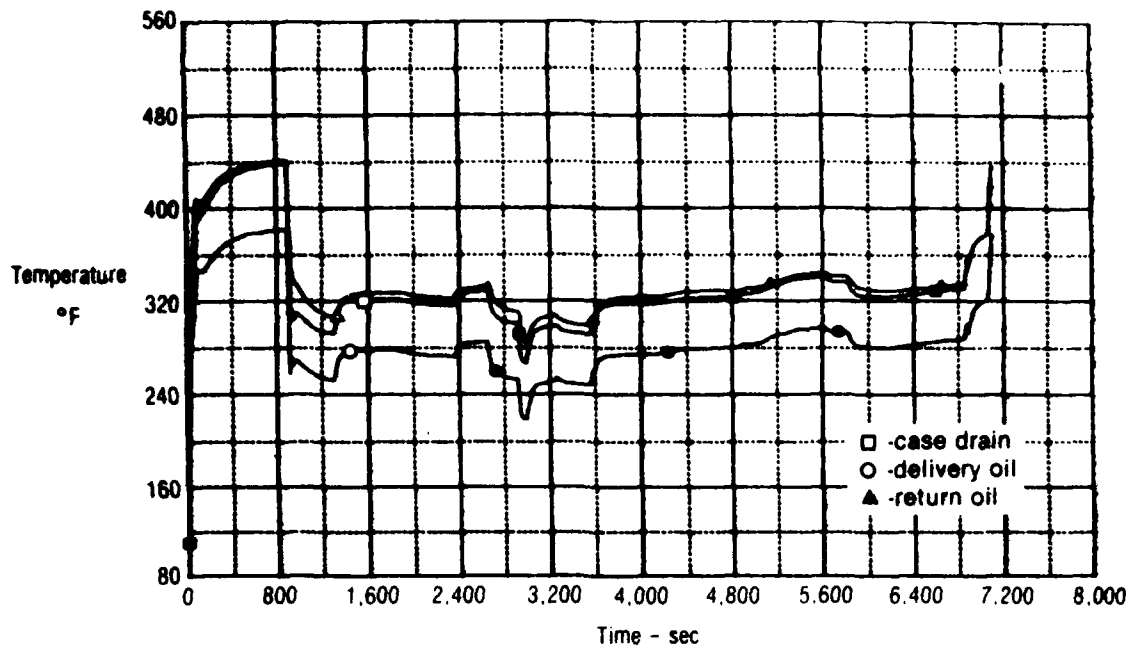


Figure 136
8,000 psi Baseline Without Ram Air HX
Utility Pump and Return Temperatures

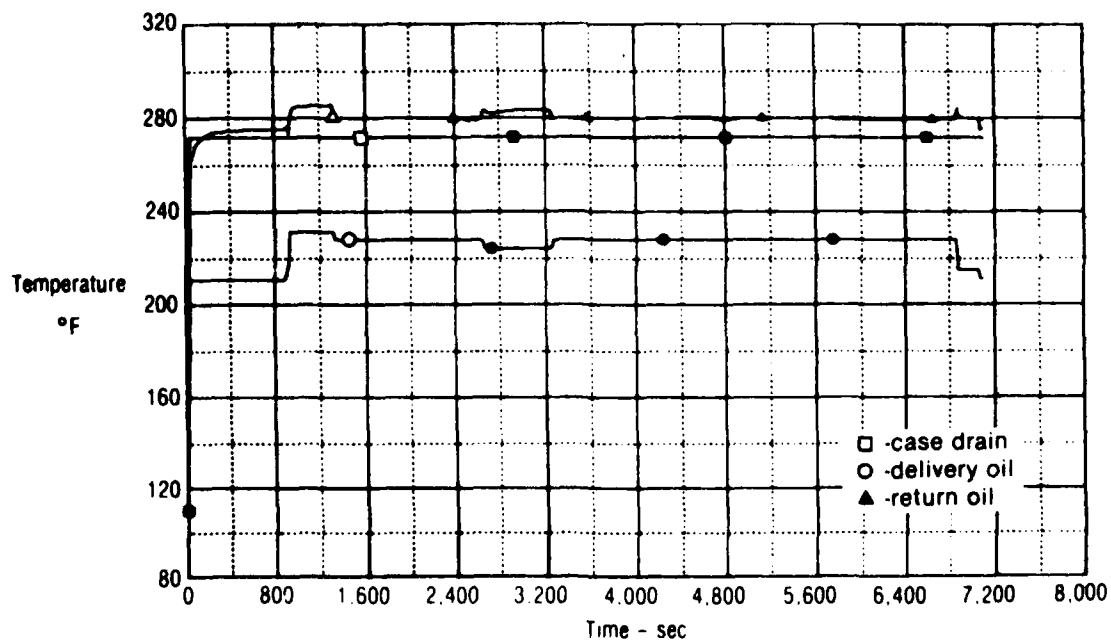


Figure 137
8,000 psi Baseline
Utility Pump and Return Temperatures

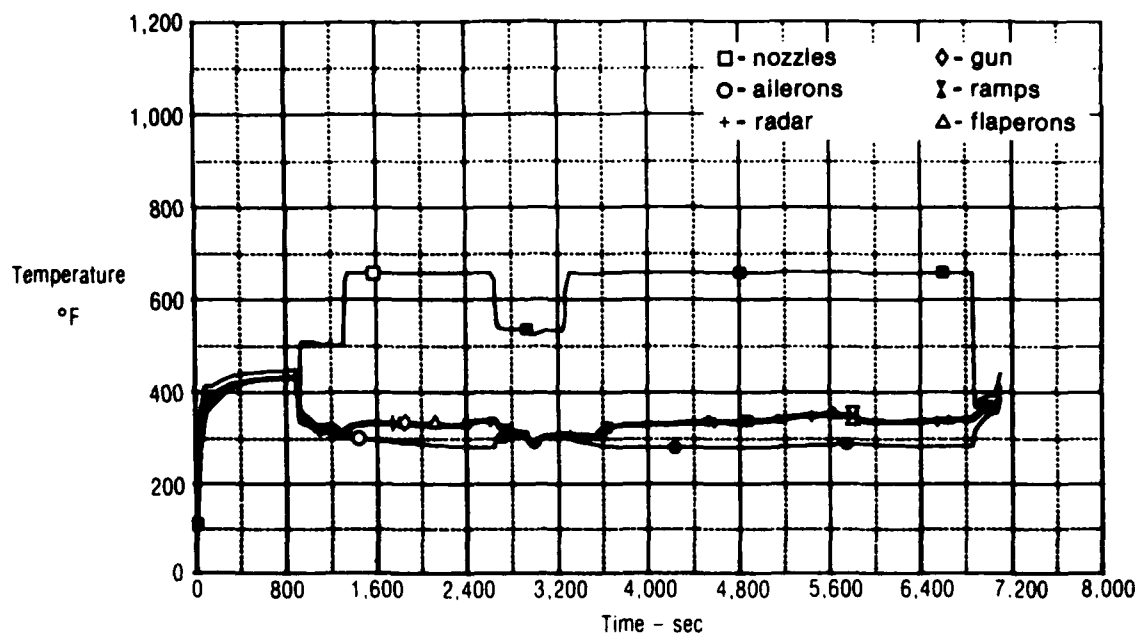


Figure 138
8,000 psi Baseline Without Ram Air HX
 Utility Actuator Exit Temperatures

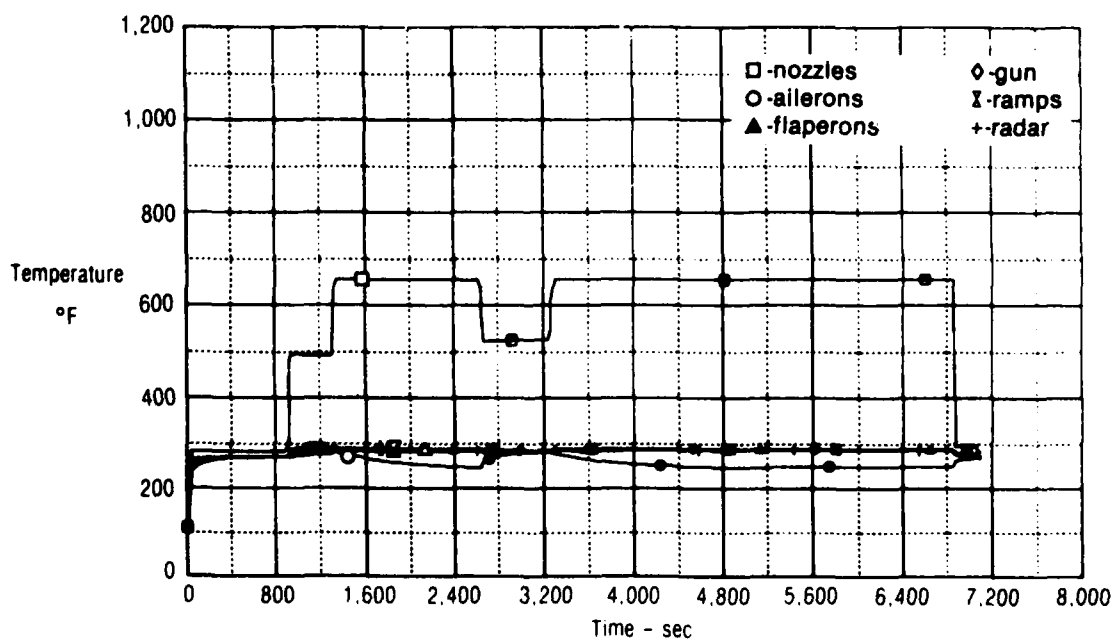


Figure 139
8,000 psi Baseline
 Utility Actuator Exit Temperatures

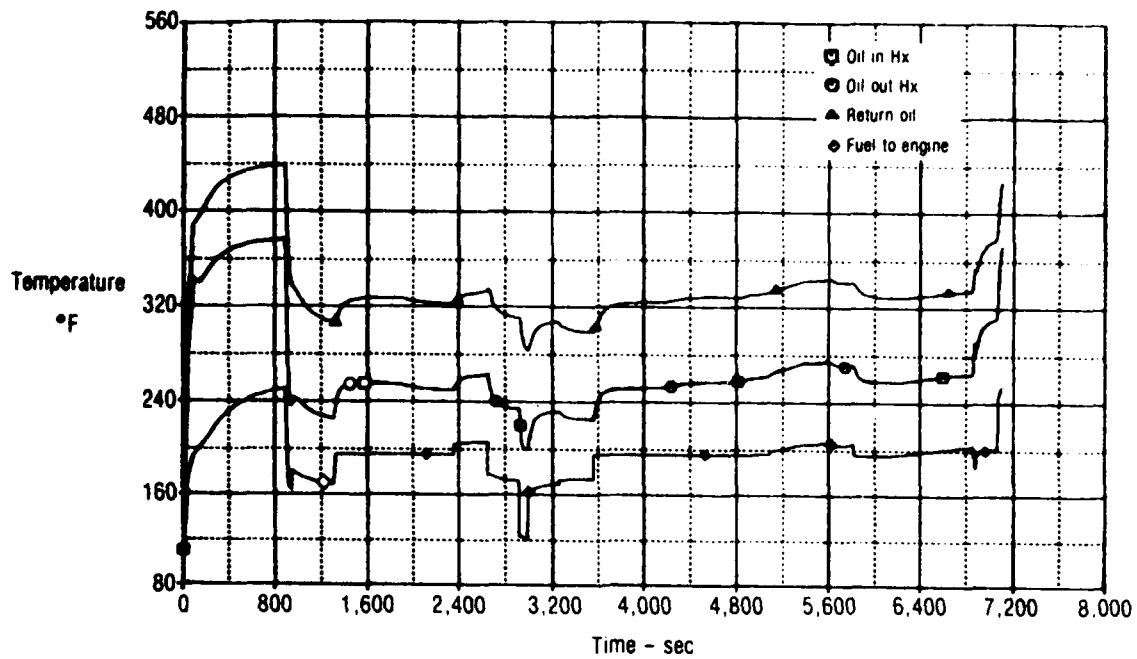


Figure 140
8,000 psi Baseline Without Ram Air HX
 Utility Ram Air HX Requirements

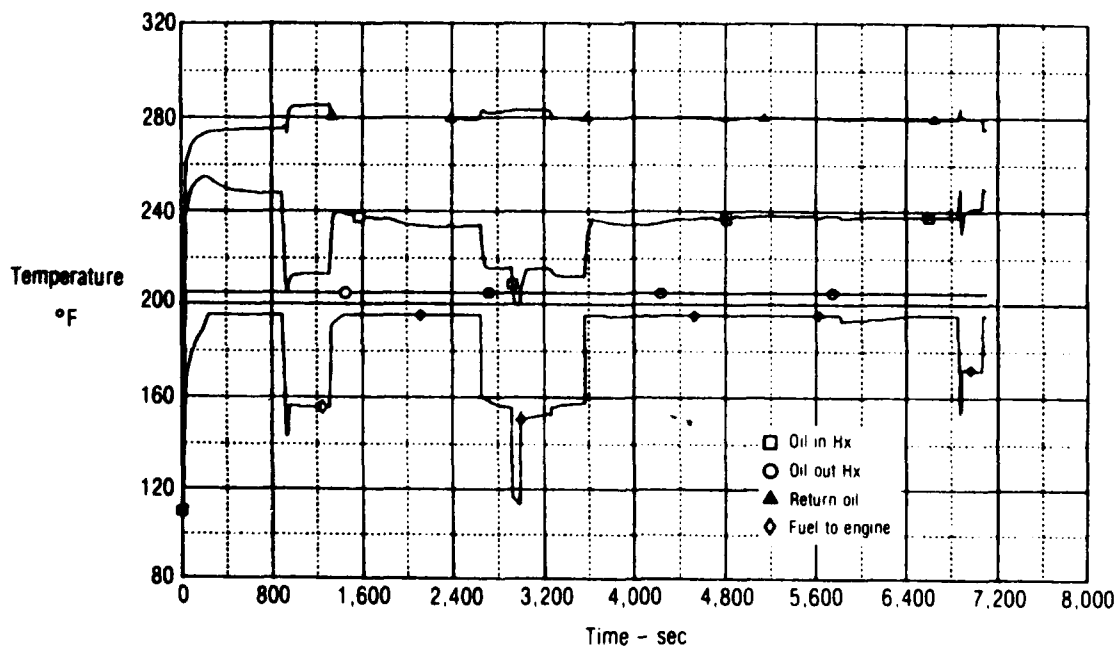


Figure 141
8,000 psi Baseline
 Utility Ram Air HX Requirements

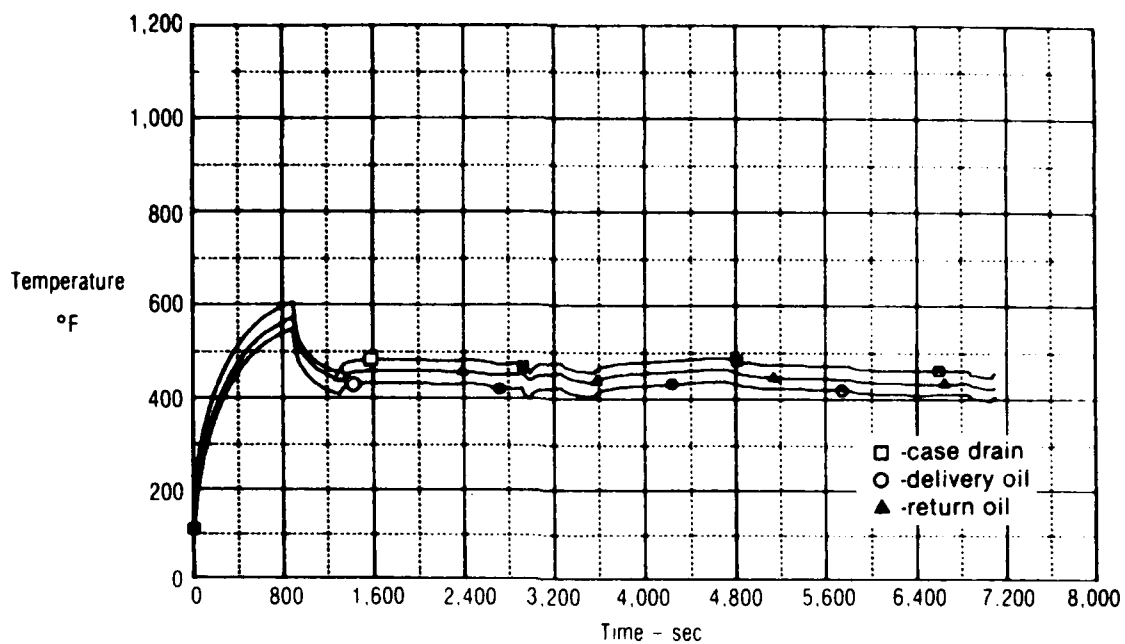


Figure 142
8,000 psi Baseline Without Ram Air HX
 PC-1 Pump and Return Temperatures

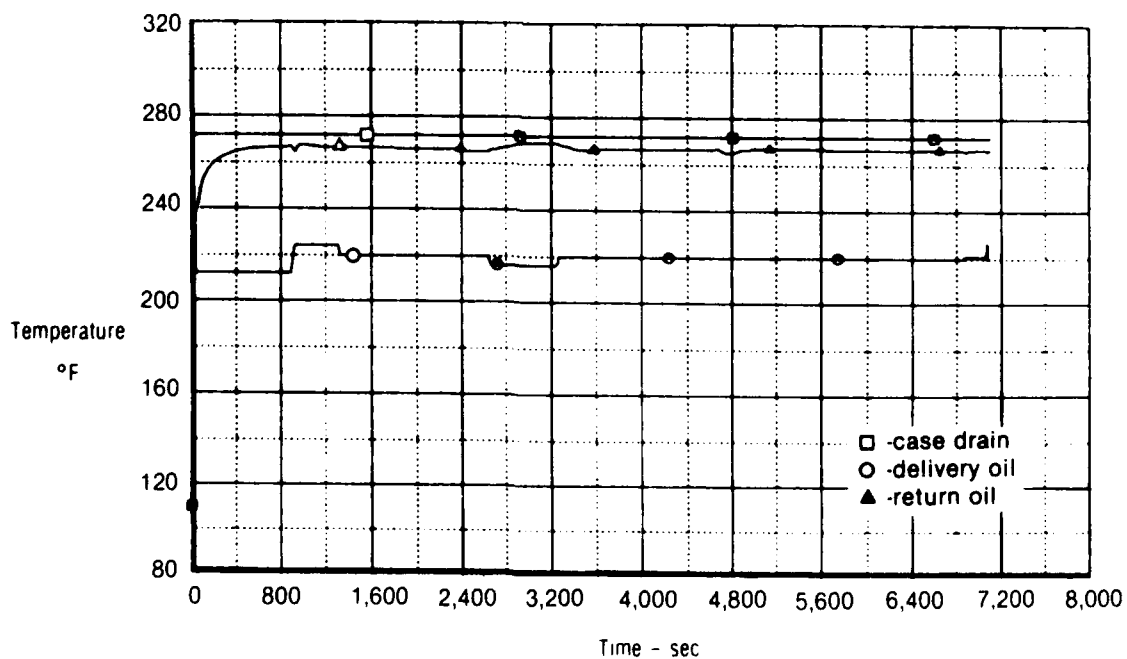


Figure 143
8,000 psi Baseline
 PC-1 Pump and Return Temperatures

GP83-0510-96

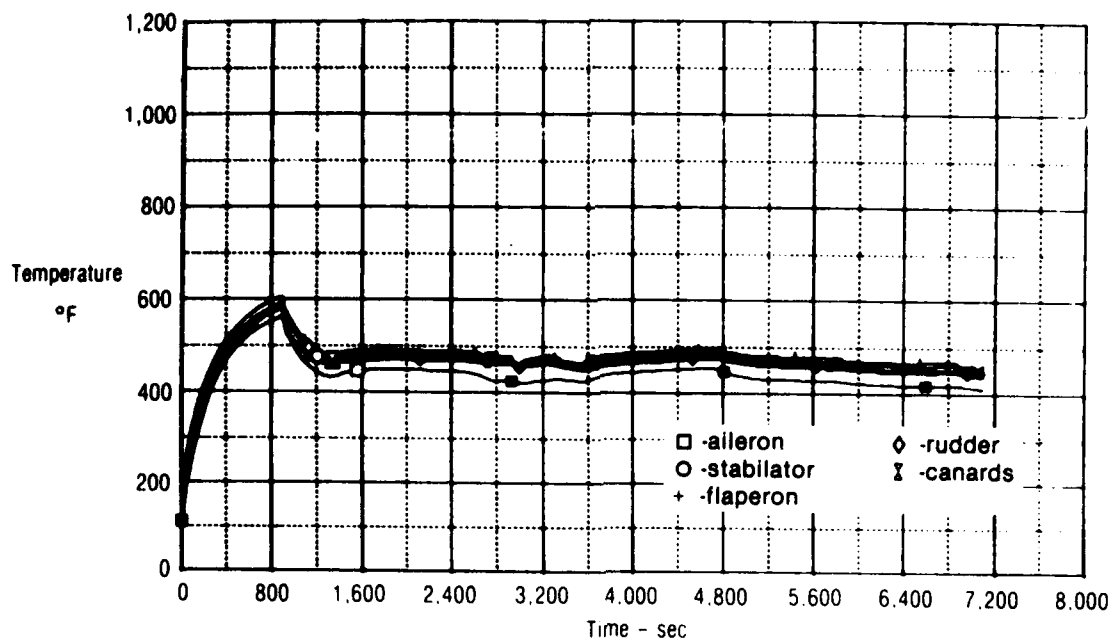


Figure 144
8,000 psi Baseline Without Ram Air HX
PC-1 Actuator Exit Temperatures

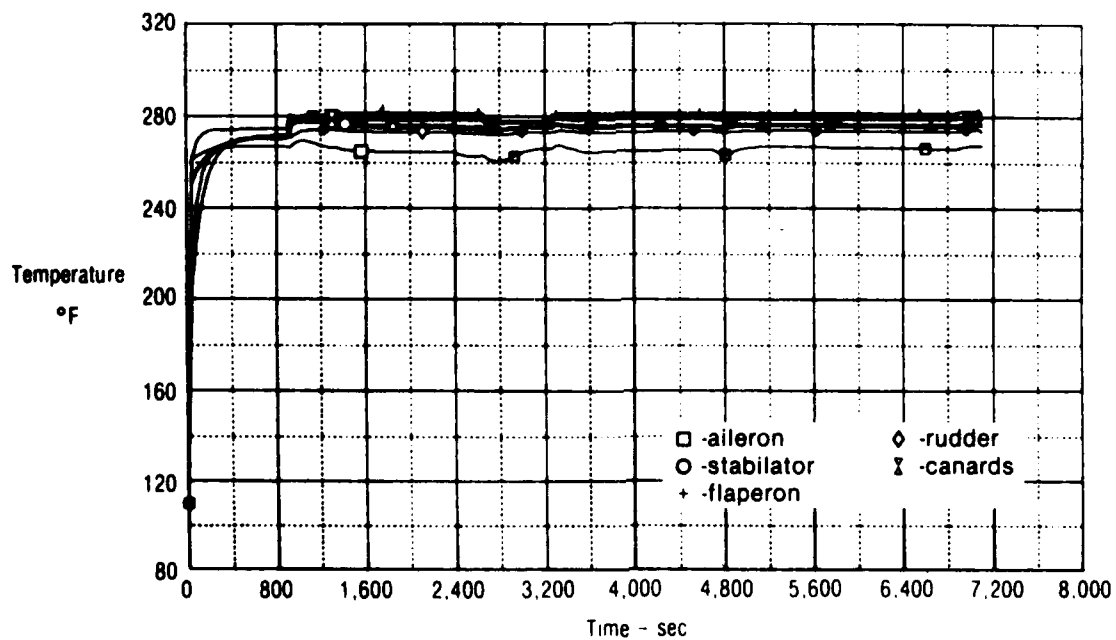


Figure 145
8,000 psi Baseline
PC-1 Actuator Exit Temperatures

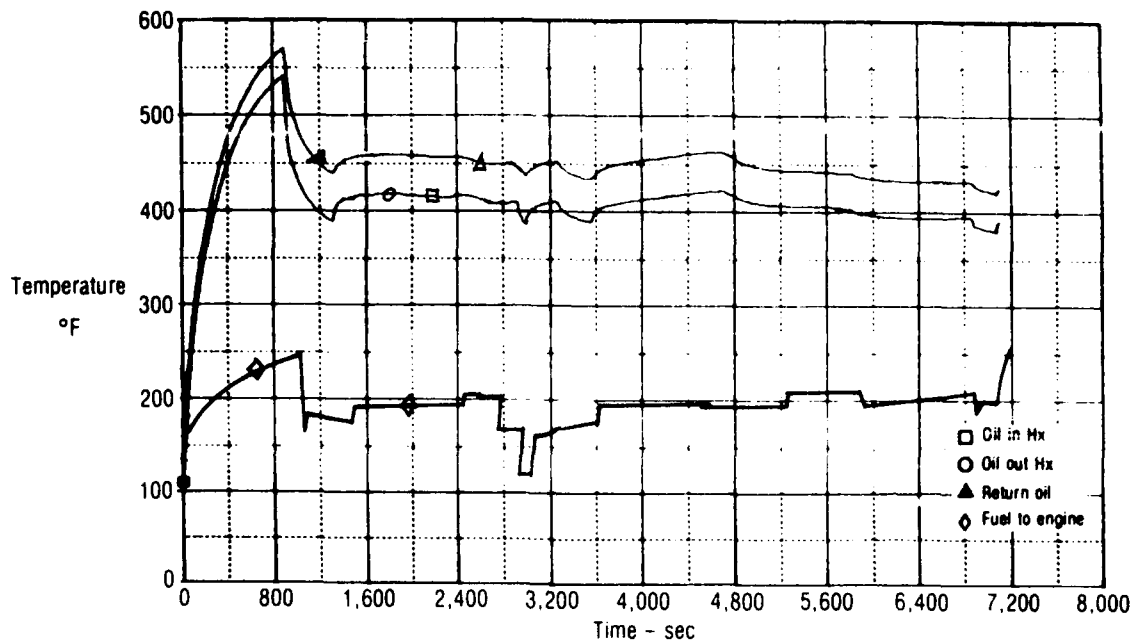


Figure 146
8,000 psi Baseline Without Ram Air HX
 PC-1 Ram Air HX Requirements

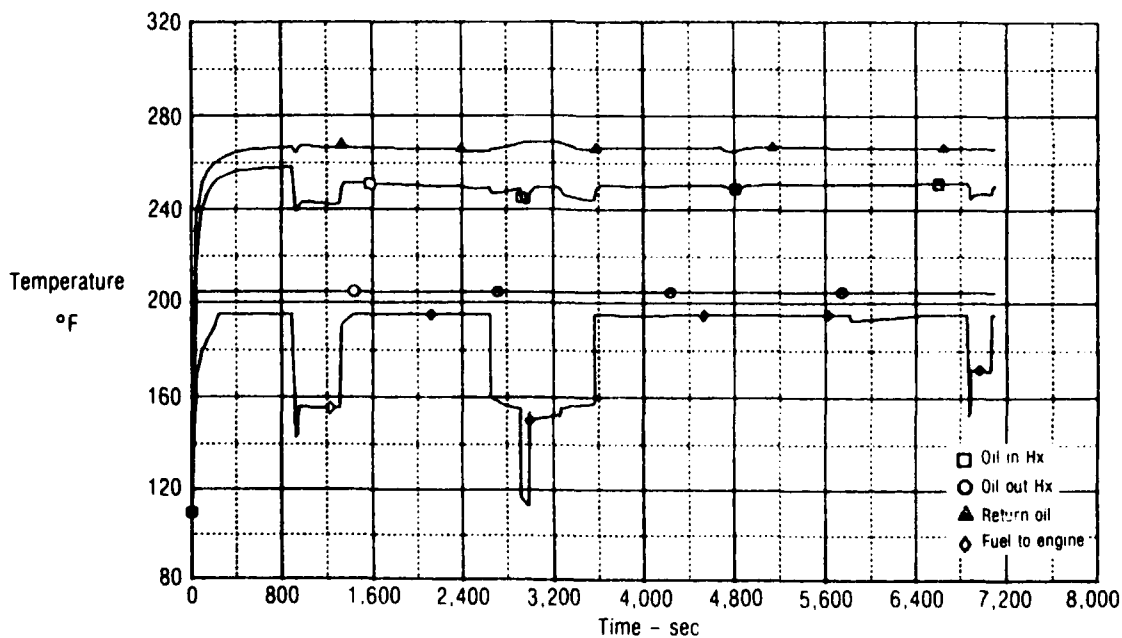


Figure 147
8,000 psi Baseline
 PC-1 Ram Air HX Requirements

After the ram air heat exchanger requirement was assessed, an analysis was derived to determine the weight impact of the ram air heat exchanger and the miscellaneous installation hardware. The ram air heat exchanger design parameters are shown in Figure 148. The heat exchangers for the baseline configurations are designed for the ground idle condition. The ground idle condition, as shown in Figures 129, 135, 141 and 147, depict the maximum heat generation level in the aircraft hydraulic system. A large fan and electric motor were sized and weighed to provide cooling capacity for ground taxiing before and after flight. As shown in Figure 149, the ram air heat exchanger weights for the 3,000 and 8,000 psi baselines are significantly different.

b. Distribution System and Component Weight - The hydraulic system weight is comprised of the categories shown in Figure 150. The only difference in the 3,000 psi weights is the CTFE fluid, which increased the weight by 241.12 lb. The 8,000 psi system is 248.96 lb lighter than the 3,000 psi system.

Design Condition

- Ground Idle
 - 8,000 psi Baseline
 - Flow Augmentation
 - Overlap Valve
 - Dry Sump
- Combat
 - Intelligent Pump
 - Combination of Concepts

Sizing Parameters

- Oil to Pump Limited to 205°F Maximum
- 1.5 psi Maximum Air Pressure Drop in Ram Air Circuit
- 25% Weight Increase for Ram Air Hx Installation
- F-15E Fuel/Oil Heat Exchangers (18.2 lb) Used in Thermal Model

Figure 148
Ram Air Heat Exchanger Design Criteria
for 8,000 psi Baseline and Concepts

Concept	Sys	HX	Fan	Motor	Duct	Install	Total	Total HX	ΔWeight
3,000 psi Baseline	PC-1 + PC-2	2.9	6.0	12.6	4.0	6.4	31.9	67.5	- 106.9
	Utility	4.3	6.5	13.9	3.8	7.1	35.6		
8,000 psi Baseline	PC-1 + PC-2	13.4	16.4	35.4	9.0	18.6	92.8	174.4	—
	Utility	13.7	14.2	29.9	7.5	16.3	81.6		

Fuel to oil heat exchanger weighs 9.1 lb per aircraft engine

Figure 149
8,000 Psi Baseline
 Ram Air Heat Exchanger Weight Breakdown - Lb

Equipment	3,000 psi		8,000 psi
	MIL-H-83282	CTFE	CTFE
Flight Control Actuators	406.00	406.00	325.73
Engine Nozzle Actuators	300.00	300.00	284.36
Utility Actuators	147.30	147.30	115.40
Heat Exchangers	85.70	85.70	192.60
Miscellaneous Components	450.00	450.00	436.25
Distribution System	374.80	374.80	197.92
Fluid	219.19	460.31	181.77
Total	1,982.99	2,224.11	1,734.03
ΔWeight	- 241.12	—	- 490.08

Figure 150
 Hydraulic System Baseline Weight - LB

The ram air heat exchanger added 106.9 lb because of the additional heat load, as mentioned in Section 3.3.1.a. The ram air heat exchanger weights, as indicated earlier, include 9.1 lb of fuel/oil heat exchanger per engine or 18.2 lb for the aircraft. All other weight categories exhibited a decrease with the implementation of 8,000 psi. Figure 151 shows the detailed weight breakdown for the baseline 8,000 psi CTFE fluid hydraulic system. Each component is divided into categories of dry weight, fluid volume, fluid weight and wet weight. The distribution system weight and the ram air heat exchanger weights are also shown.

After the weight impact was determined for the baseline hydraulic configurations, the aircraft was resized by growth factor analysis to determine the total aircraft weight. The weight was then distributed to structure, fuel, engines and subsystems. The resulting weight breakdowns are shown in Figure 152.

	Dry Weight (lb)	Fluid Volume (in. ³)	Fluid Weight (lb) CTFE	Wet Weight (lb)
Jet Fuel Starter Valve				
JFS Manifold	6.52			6.52
Other	1.28			1.28
Miscellaneous				
JFS Hand Pump	3.00			3.00
Accumulator	35.98	111.88	7.16	43.14
Other	2.47	38.88	2.36	4.83
Subtotal	49.25	148.76	9.52	58.77
Aerial Refuel Receptacle Actuator	1.48	1.88	0.12	1.58
Valve	3.06			3.06
Subtotal	4.52	1.88	0.12	4.64
Air Induction Actuators				
Bypass Door	12.94	20.63	1.32	14.26
Diffuser Ramp	29.27	75.63	4.84	34.11
First Ramp	21.28	26.25	1.88	22.96
Other	2.04	1.09	0.07	2.11
Subtotal	65.53	123.60	7.91	73.44
Nozzle Controls Actuators				
Upper Rotating Vane	24.20	7.45	0.48	24.63
Lower Rotating Vane	24.20	7.45	0.48	24.68
Outboard Divergent Flap	84.52	56.04	3.59	88.11
Lower Divergent Flap	84.52	56.04	3.59	88.11
Convergent Flap	66.92	60.68	3.88	70.80
Valves	9.66	7.81	0.50	10.16
Subtotal	294.02	195.47	12.52	306.54
Canards Actuators	90.50	85.00	5.14	95.94
Valves	9.66	7.81	0.50	10.16
Subtotal	100.16	92.81	5.94	106.10
Ailerons Actuators	53.00	9.00	0.58	53.58
Valves				
Switching	9.66	7.81	0.50	10.16
Other	0.58			0.58
Subtotal	63.24	16.81	1.08	64.32
Stabilator Actuators	90.50	85.00	5.44	95.94
Valves				
Switching	9.66	7.81	0.50	10.16
Other	2.08			2.08
Subtotal	102.24	92.81	5.94	108.18
Arresting Hook Uplatch Actuator	2.29	5.63	0.36	2.65
Valve	0.99			0.99
Other	0.30			0.30
Subtotal	3.58	5.63	0.36	3.94
Main Landing Gear Actuator	20.74	65.00	4.16	24.90
Retract	6.64	6.72	0.43	7.07
Uplock				
Valves				
Uplock	0.45			0.45
Retract	12.51			12.51
Brake Operate	5.70			5.70
Emergency Ext	1.50			1.50
Miscellaneous	5.70	15.16	0.97	6.67
Subtotal	53.24	86.88	5.56	58.80
Ram Air Heat Exchanger Requirements				
	PC System	Utility System	Total	
Fuel/Oil HX	9.1	9.1	18.2	
Ram Air HX	13.4	13.7	27.1	
Fans	18.4	14.2	30.6	
Electric Motor	35.4	29.9	65.3	
Duct	9.0	7.5	16.5	
Installation	18.6	16.3	34.9	
Total	101.9	90.7	192.6	

	Dry Weight (lb)	Fluid Volume (in. ³)	Fluid Weight (lb) CTFE	Wet Weight (lb)
Nose Landing Gear Actuator				
Uplock	3.18	2.66	0.17	3.35
Retract	8.16	25.31	1.62	9.78
Steering and Damper Valves	19.80			19.80
Miscellaneous	0.30			0.30
Other	6.70			6.70
	1.17			1.17
	1.17			1.17
Miscellaneous	0.20			0.20
Other	0.90	6.09	0.39	1.29
Subtotal	41.58	34.06	2.18	43.76
Emergency Generator Valve	4.32	4.06	0.26	4.58
Subtotal	4.32	4.06	0.26	4.58
Gun System Valve				
Gun Flow Regulator	2.34	6.72	0.43	2.77
Other	0.10			0.10
Subtotal	2.44	6.72	0.43	2.87
Rudder Actuator	38.73	15.63	1.00	39.73
Other	3.00			3.00
Subtotal	41.73	15.63	1.00	42.73
Flaperon Actuator	53.00	9.00	0.58	53.58
Subtotal	53.00	9.00	0.58	53.58
ECS Auxiliary Air Inlet Actuator	2.94	1.88	0.12	3.06
Valve	1.26			1.26
Subtotal	4.20	1.88	0.12	4.32
Hydraulic Utility System Valve				
Temperature Regulator	2.04	6.25	0.40	2.44
Other	4.50			4.50
Miscellaneous				
Pump	64.00	80.00	5	69.12
Reservoir	22.90	348.00	22.27	45.17
Primary Heat Exchanger	0.83	0.63	0.04	0.87
Primary HX Valve	1.17			1.17
Other	41.73	43.44	2.78	44.51
Subtotal	137.17	478.32	30.61	167.78
Canopy Actuator				
Main	5.04	8.44	0.54	5.58
Lock	1.46	2.03	0.13	1.59
Other	0.35		0.35	
Valve	3.33			3.33
Miscellaneous				
Accumulator	3.47	14.22	0.91	4.38
Other	0.35			0.35
Subtotal	14.00	24.69	1.58	15.58
Hydraulic PC-1 and PC-2 Valve				
Temperature Regulator	4.08	12.50	0.80	4.68
Other	0.40			0.40
Miscellaneous				
Pump	64.00	80.00	5.12	69.12
Reservoir	18.60	224.00	14.34	32.94
Other	40.44	25.00	1.60	42.04
Subtotal	127.52	341.50	21.86	149.38
Total	1,181.74	1,880.51	107.57	1,289.31
Distribution System				
	Dry Weight (lb)	Fluid Volume (in. ³)	Fluid Weight (lb)	Wet Weight (lb)
PC-1 and PC-2	92.66	504.54	32.29	124.95
Utility	105.26	654.89	41.91	147.17
Total	197.92	1,159.43	74.20	272.12

Figure 151
8,000 psi Baseline
Hydraulic Equipment Weight

	3,000 psi MIL-H-83282		3,000 psi CTFE				8,000 psi CTFE			
	Total Weight	Hydraulic Weight	Total Weight	Hydraulic Contri- bution	Δ Weight (1)	ΔHydraulic Contri- bution(1)	Total Weight	Hydraulic Contri- bution	Δ Weight (1)	ΔHydraulic Contri- bution(1)
Airframe	14,438	147.82	14,767	151.46	+329	+3.64	14,123	144.85	-315	-2.97
Engine	6,321	—	6,422	—	+121	—	6,208	—	-113	—
Avionics	1,862	—	1,869	—	+7	—	1,855	—	-7	—
Subsystems	5,939	1,835.17	6,021	2,072.65	+82	+237.48	5,861	1,589.18	-78	-245.99
Total Empty Wt	28,560	1,982.99	29,099	2,244.11	+539	+241.12	28,047	1,734.03	-513	-248.96
Fuel	13,925	—	14,169	—	+244	—	13,728	—	-197	—
Payload	2,040	—	2,040	—	—	—	2,040	—	—	—
Oxygen	28	—	28	—	—	—	28	—	—	—
Crew	215	—	215	—	—	—	215	—	—	—
Unusable Fuel	493	—	493	—	—	—	493	—	—	—
Oil	76	—	76	—	—	—	76	—	—	—
Gun/Ammo	783	—	783	—	—	—	783	—	—	—
Misc Equip	50	—	50	—	—	—	50	—	—	—
Gross Weight	46,170	1,982.99	46,953	2,244.11	+783	+241.12	45,460	1,734.03	-710	-248.96

(1) Delta weights are relative to the 3,000 psi MIL-H-83282 baseline

Figure 152
F-15 SMTD Baseline Aircraft Weight Summary - LB

c. Life Cycle Costs - Life cycle costs were developed on a hydraulic and related equipment basis, and on a total aircraft basis. The procedure for calculating these costs is described in Section 3.1.2.b.

The baseline LCC study was established at 3,000 psi with MIL-H-83282. LCCs were then computed for the 3,000 and 8,000 psi baselines with CTFE fluid. Prior to performing the analysis, unit procurement costs were derived for the cost categories shown in Figure 153. The 8,000 psi baseline resulted in a 20-25 percent increase in component procurement costs. The 8,000 psi system yielded \$138,000 per aircraft increase over the 3,000 psi baseline. However, there was a significant decrease in the distribution system cost/weight. This was because at 8,000 psi, the lines can be sized smaller to attain the desired actuator rates. The 3,000 psi CTFE baseline was derived on a drain and fill basis with the fluid. This resulted in a \$3,800 increase in unit procurement cost.

Figure 154 summarizes the hydraulic and aircraft system LCC. As shown, the hydraulic systems total LCCs for the 8,000 psi CTFE hydraulic system, was \$96 M higher than the 3,000 psi MIL-H-83282 baseline, while the 3,000 psi CTFE baseline LCC relative to 3,000 psi MIL-H-83282 baseline, resulted in a \$2 M increase. Alternately, the aircraft LCCs for the 8,000 psi CTFE system was \$112 M less than the 3,000 psi MIL-H-83282 baseline. The 3,000 psi CTFE baseline LCC relative to the 3,000 psi MIL-H-83282 baseline, increased by \$203 M.

Cost Category	3,000 psi		8,000 psi
	MIL-H-83282	CTFE	CTFE
Flight Control Actuators	—	—	+ 29.0
Engine Nozzle Actuators	—	—	+ 55.0
Utility Actuators	—	—	+ 18.0
Heat Exchangers	—	—	+ 47.0
Miscellaneous Components	—	—	+ 25.0
Distribution System	—	—	- 45.0
Fluid	—	+ 3.8	+ 1.6
Integration and Test	—	—	+ 8.0
Total	—	+ 3.8	+ 138.6

Figure 153
Baseline Hydraulic Unit (Shipset) Procurement Cost
 ΔThousands of FY 85 Dollars

	3,000 psi MIL-H-83282		3,000 psi CTFE		8,000 psi CTFE	
	Hydraulic ⁽¹⁾	Total ⁽²⁾	Hydraulic ⁽¹⁾	Total ⁽²⁾	Hydraulic ⁽¹⁾	Total ⁽²⁾
Acquisition						
Development	—	—	0	0	+ 22	+ 9
Investment	—	—	+ 2	+ 2	+ 70	+ 37
Flyaway	—	—	+ 2	+ 2	+ 69	+ 44
Other	—	—	0	0	+ 1	- 7
Subtotal	—	—	+ 2	+ 2	+ 92	+ 46
15 Years Operation and Support	—	—	0	+ 147	+ 4	- 107
Fuel	—	—	0	+ 54	0	- 51
Total LCC	—	—	+ 2	+ 203	+ 96	- 112

Notes:

- (1) Hydraulics and related equipment
 (2) Total aircraft costs, including hydraulics

Figure 154
F-15 S/MTD Aircraft Life Cycle Cost Summary
 ΔMillions of FY 85 Dollars

Comparing the hydraulic and aircraft systems for the same pressure and fluid requires explanation. The 3,000 psi CTFE system cost comparisons were:

- o Development costs remained the same because only the fluid changed
- o Investment costs increased \$2 M because of the increased CTFE fluid costs
- o Flyaway costs increased \$2 M as reflected in the \$2 M investment cost increase
- o Costs included in the other category remained unchanged
- o Operation and support costs increased \$147 M due to increased CTFE fluid weight which affected aircraft maintenance and support costs
- o Fuel costs increased \$54 M because of increased hydraulic weight due to CTFE fluid

The 8,000 psi CTFE system cost comparisons were:

- o Development costs of the hydraulic system increased \$22 M due to increased component material cost and complexity. This increase in cost was partially offset by the decrease in hydraulic weight, which resulted in an overall development cost increase of only \$9 M
- o As with development costs, hydraulic investment costs increased \$70 M while aircraft costs increased only \$37 M
- o As with development and investment costs, flyaway costs increased \$69 M, while aircraft costs increased only \$44 M
- o Other costs for the hydraulic system included \$1 M due to increased fluid costs, while total aircraft costs decreased \$7 M due to the decreased hydraulic weight
- o Operation and support for the hydraulic system increased \$4 M due to the added complexity at 8,000 psi and the use of titanium materials. The aircraft costs decreased \$107 M because of the significant decrease in hydraulic weight
- o Fuel costs decreased \$54 M because of the decrease in hydraulic weight

Comparing the two CTFE columns, there is a difference indicated of \$315 M. This shows that a significant cost savings can be achieved by designing at 8,000 psi.

3.3.2 Overlap Valves - Valve overlap, illustrated in Figure 102, reduces the level of heat rejection by decreasing the servovalve null leakage an order of magnitude. This concept minimizes the null condition pump flow requirement. Thus, less pump heat is injected into the system. This savings is significant when all actuators are considered. All valves in the hydraulic system implemented the overlap concept for this study.

a. Thermal Analysis - The thermal analysis for the 8,000 psi CTFE overlap valve concept was divided into two segments. The first segment was the pump heat rejection and the other was the average operational and average leakage flow rates.

The pump heat rejection is shown in Figure 101 for the overlap valve system. The heat rejection was the same as in the baseline 8,000 psi system, because operational flow demands were unchanged. Figure 155 shows the thermal model parameters established for the pump. The case drain flow increased to 2.7 gpm and the heat rejection to the case drain/pump discharge port changed to a 93 percent/7 percent relationship. This was based on pump heat rejection tests performed at MCAIR. At the valve null condition, less flow was directed to the discharge and more to the case drain.

Configuration	Pump Heat Rejection (hp)	H.R. Accounted for (hp)	Case Drain Flow (gpm)	H.R. to Case Drain (hp)	H.R. to Discharge (hp)	Actuator Operational Flows	Actuator Leakage Flows	System Pressure (psi)
8,000 psi Baseline	16.64	14.14 (85%)	2.0	9.47 (67%)	4.67 (33%)	Figure 122	Figure 123	8,000
0.010 Overlap Valve	16.64	14.14 (85%)	2.0	13.15 (93%)	1.0 (7%)	Figure 156	Figure 157	8,000

Figure 155
Overlap Valve Thermal Model Parameters

The actuator operational and leakage flow rates are shown in Figures 156 and 157. The operational flows are the same as the baseline, but the leakage flows decreased by a factor of ten.

Figures 158 through 169 show the temperatures that were computed using the thermal analyses computer model, as mentioned in Section 3.1.2.a. These plots should be interpreted as described in Section 3.3.1.a.

The ram air heat exchanger weight impact when utilizing the overlap valve concept, is shown in Figure 170. As shown, the total weight decreased by 53.3 lb. Note that the majority of weight was due to the fans and motors utilized during ground operation.

Per Actuator	Actuator Flow (Unloaded)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	1.08	0.228 ¹	0.034	0.034	0.023	0.164	0.023	0.023	0.076	0.011
Flaperon	1.31	0.875 ¹	—	0.137	—	0.051	—	—	0.122	0.014
Rudder	1.37	0.175 ¹	0.043	0.015	0.015	0.208	0.015	0.015	0.143	0.143
Stabilator	11.60	1.75 ¹	1.092	0.605	0.242	1.732	0.242	0.484	1.092	0.121
Canard	9.81	1.75 ¹	0.768	0.307	0.102	1.489	0.102	0.307	0.512	0.102
Utility System										
Gun	10.88	—	—	—	—	0.107	—	—	—	—
Steering	0.90	0.64	0.64	—	—	—	—	—	—	0.64
Radar	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64
Nozzles	—	7.28	5.81	0.204	0.061	0.162	0.061	0.039	1.08	0.54
Ramps	5.91	—	0.006	0.008	0.008	0.213	0.008	0.006	0.006	—
Aileron	2.17	0.456 ¹	0.069	0.069	0.046	0.329	0.046	0.046	0.152	0.022
Flaperon	2.61	1.75 ¹	—	0.274	—	0.099	—	—	0.244	0.028

¹ = leakage

Note: Flows are mean values over flight phase in GPM

Figure 156
F-15 SMTD
 Operational Flow 8,000 psi 0.010 Overlap Valve

Per Actuator	Null Leakage (per Aircraft)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
		PC-1 and PC-2 Systems								
Aileron	0.013	0.013	0.0092	0.0092	0.0104	—	0.0104	0.0104	0.0042	0.0118
Flaperon	0.05	0.05	0.05	—	0.05	0.05	0.05	0.05	—	0.045
Rudder	0.01	0.01	0.01	0.007	0.009	—	0.009	0.009	—	—
Stabilator	0.10	0.10	0.01	0.05	0.08	0.02	0.08	0.06	0.01	0.09
Canard	0.10	0.10	0.025	0.07	0.09	0.08	0.09	0.07	0.05	0.09
		Utility System								
Gear	0.034	0.034	0.034	0.034	0.034	—	0.034	0.034	0.034	0.034
Steering	0.011	0.0032	0.0032	—	—	—	—	—	—	0.0032
Radar	—	—	—	—	—	—	—	—	—	—
Nozzles	0.416	0.416	—	—	—	—	—	0.208	0.3786	0.416
Ramps	0.156	0.156	0.155	0.1528	0.1528	0.1326	0.1528	0.155	0.155	0.144
Aileron	0.026	0.026	0.0182	0.0208	—	—	—	0.0208	0.0086	0.0234
Flaperon	0.10	0.10	0.10	—	0.10	0.10	0.10	0.10	—	0.09

Note. Flows are mean values over flight phase in GPM

Figure 157
F-15 SMTD
 Leakage Flow 8,000 psi 0.010 Overlap Valve

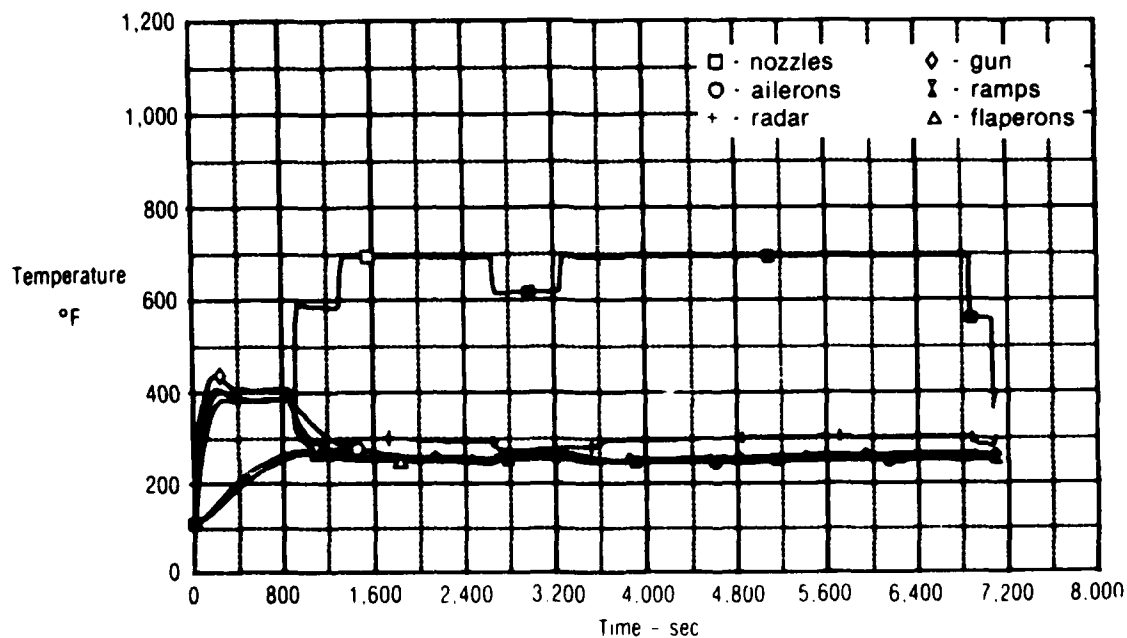


Figure 158
 8,000 psi Overlap Valve Without Ram Air HX
 Utility Actuator Exit Temperatures

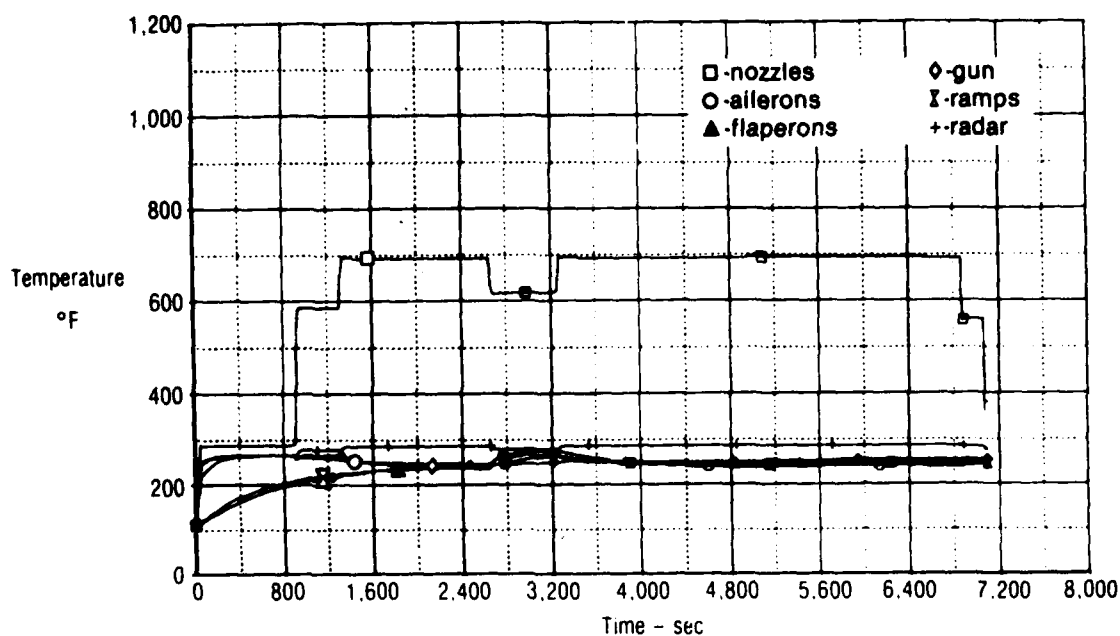


Figure 159
8,000 psi Overlap Valve
 Utility Actuator Exit Temperatures

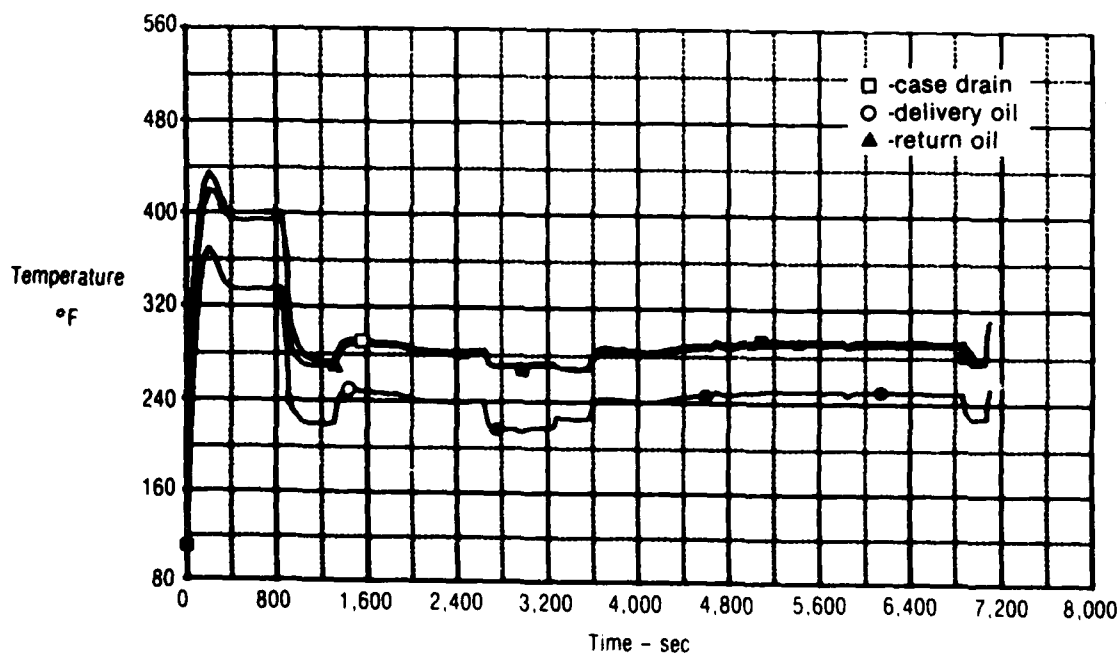


Figure 160
8,000 psi Overlap Valve Without Ram Air HX
 Utility Pump and Return Temperatures

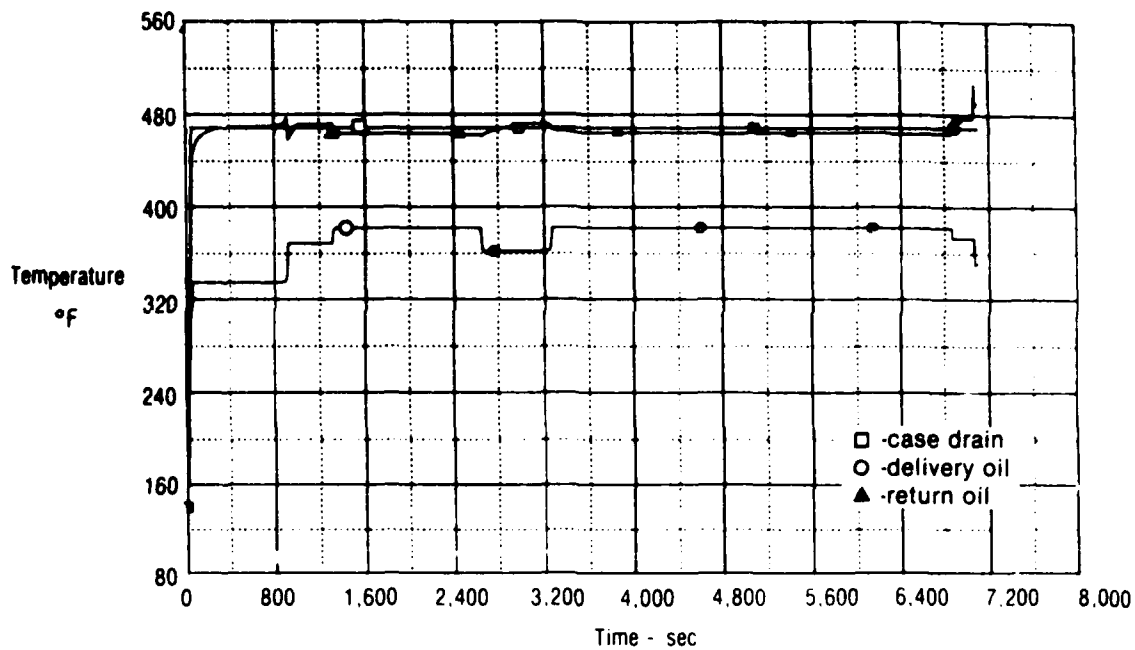


Figure 161
8,000 psi Overlap Valve
 Utility Pump and Return Temperatures

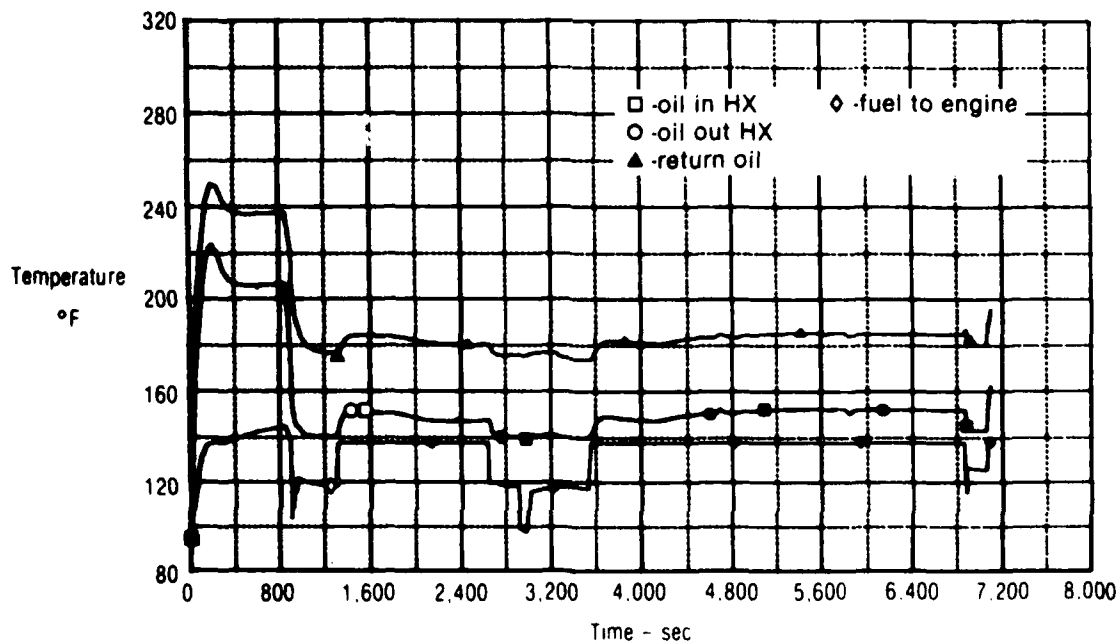


Figure 162
8,000 psi Overlap Valve Without Ram Air HX
 Utility Ram Air HX Requirements

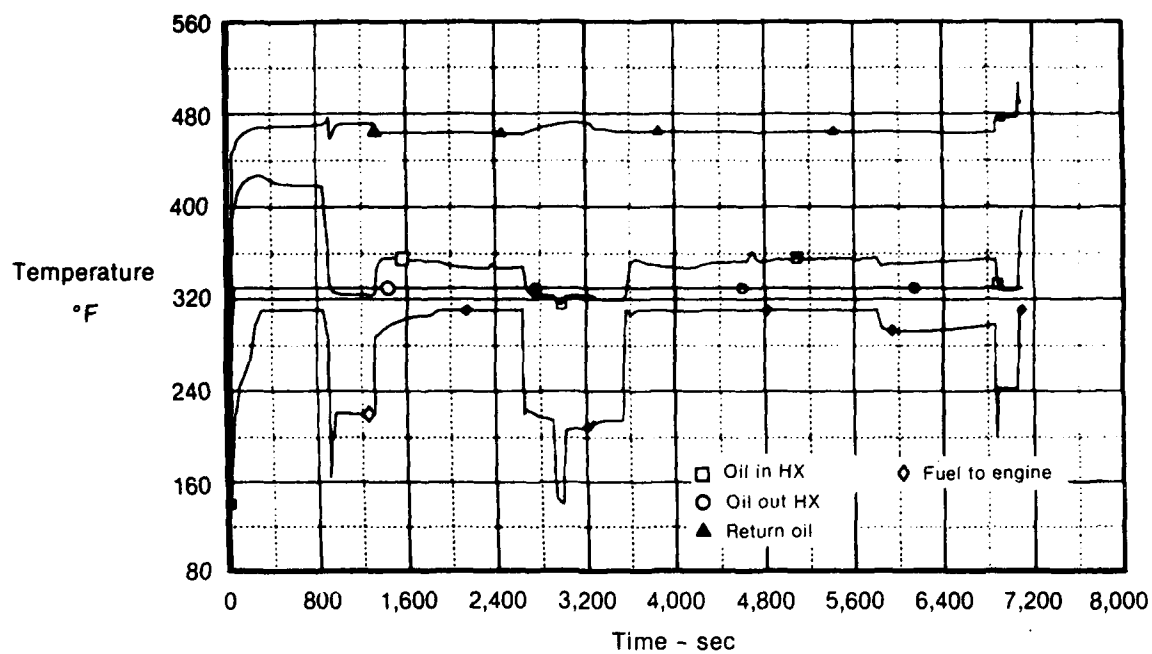


Figure 163
8,000 psi Overlap Valve
 Utility Ram Air HX Requirements

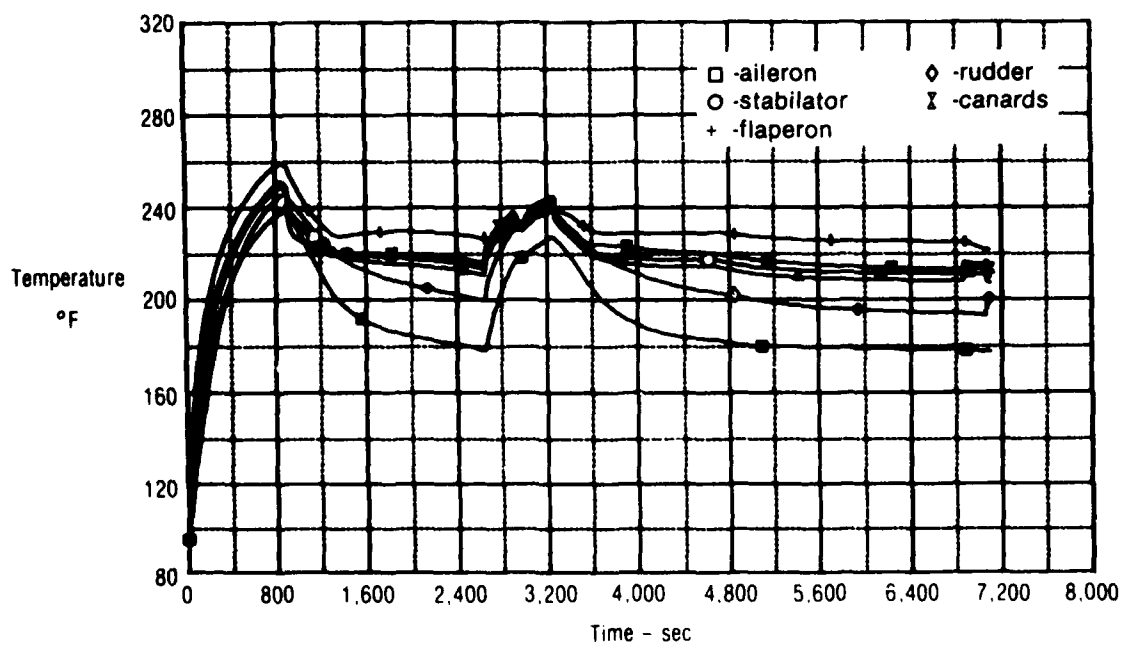


Figure 164
8,000 psi Overlap Valve Without Ram Air HX
 PC-1 Actuator Exit Temperatures

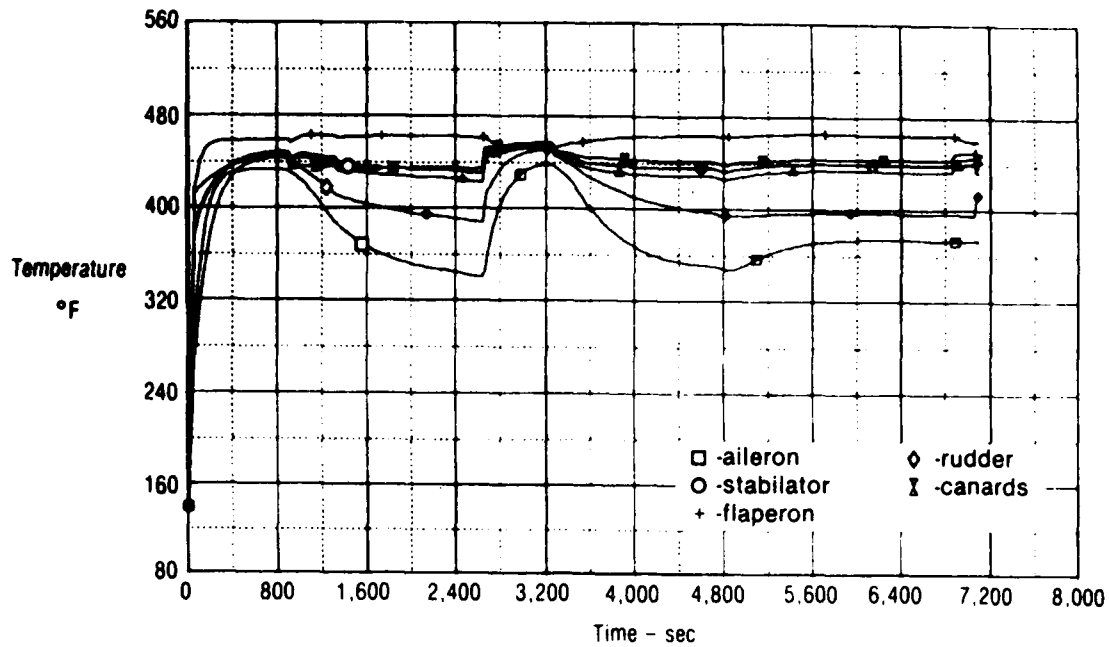


Figure 165
8,000 psi Overlap Valve
 PC-1 Actuator Exit Temperatures

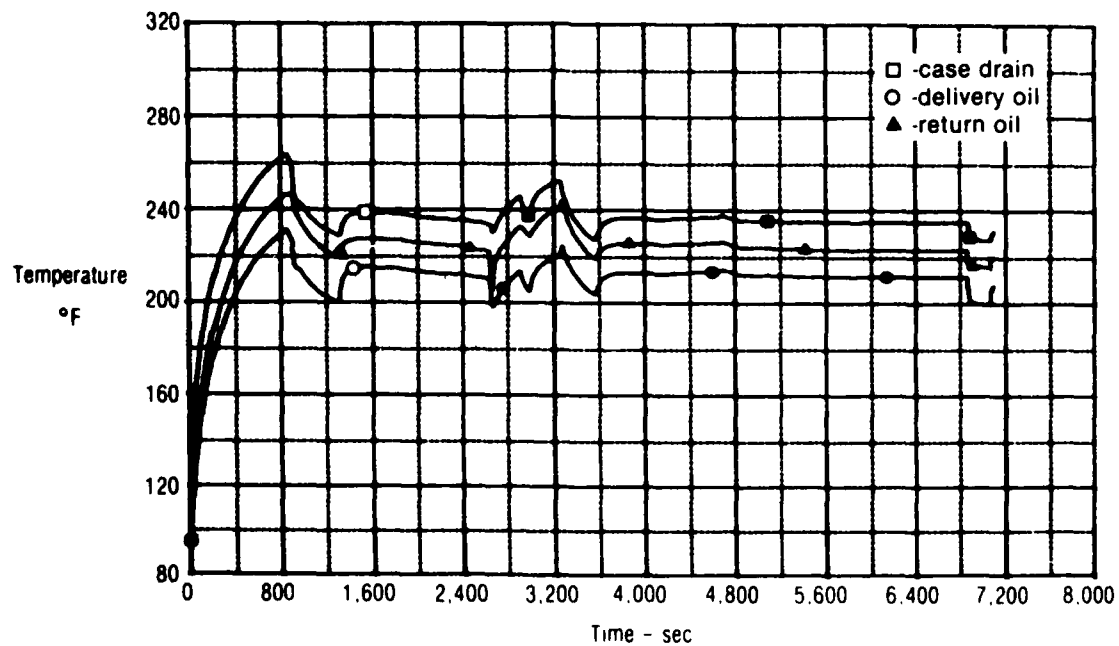


Figure 166
8,000 psi Overlap Valve Without Ram Air HX
 PC-1 Pump and Return Temperatures

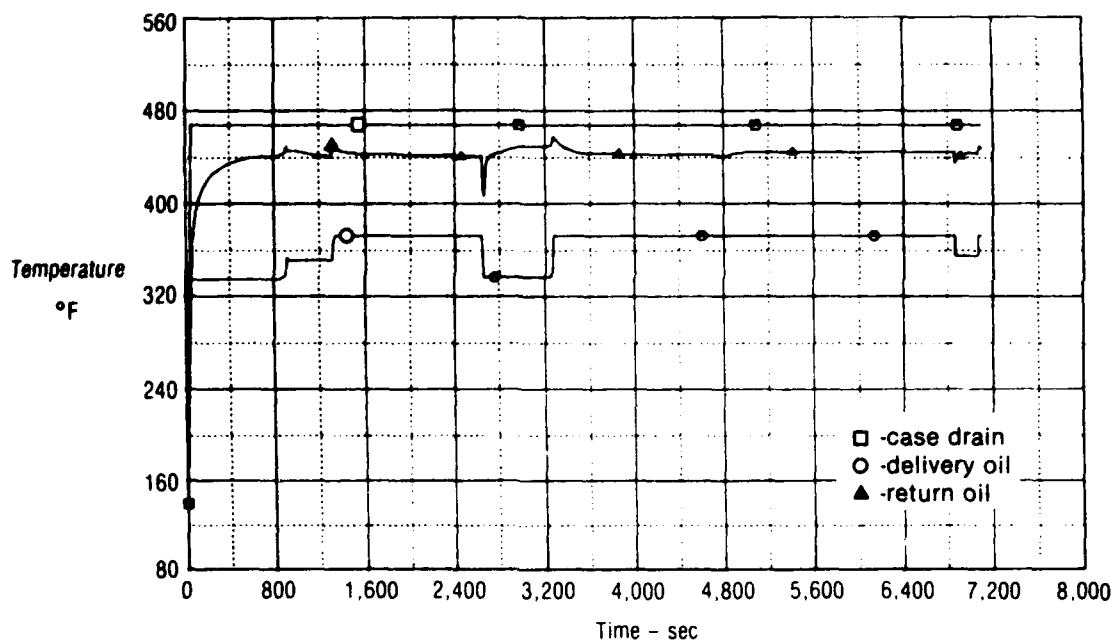


Figure 167
8,000 psi Overlap Valve
PC-1 Pump and Return Temperatures

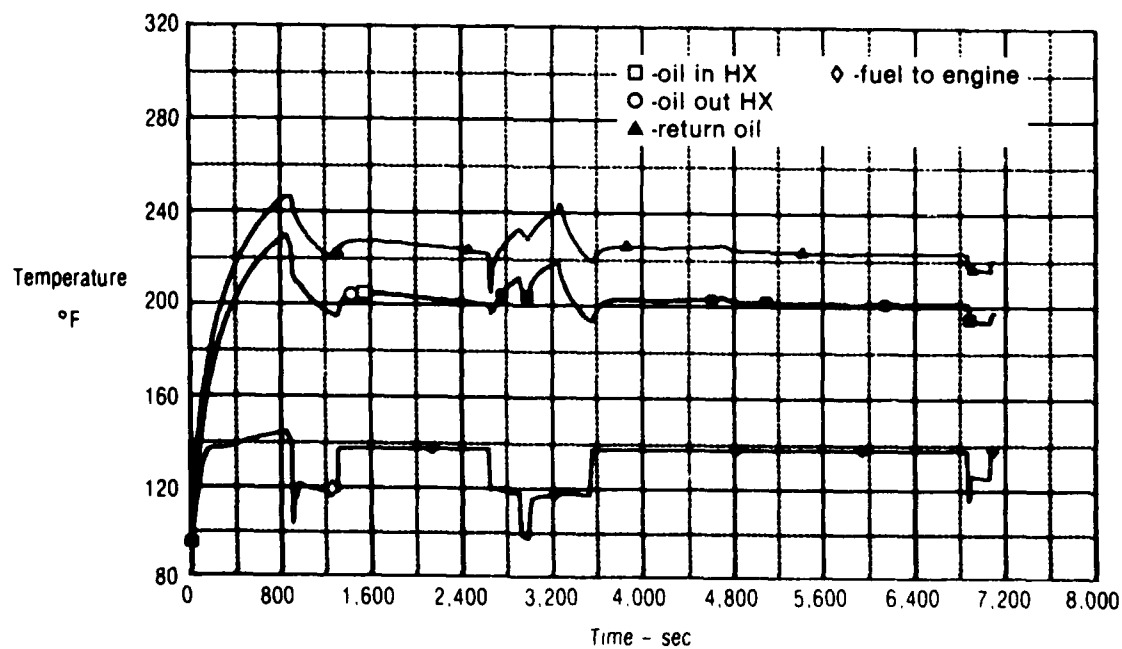


Figure 168
8,000 psi Overlap Valve Without Ram Air HX
PC-1 Ram Air HX Requirements

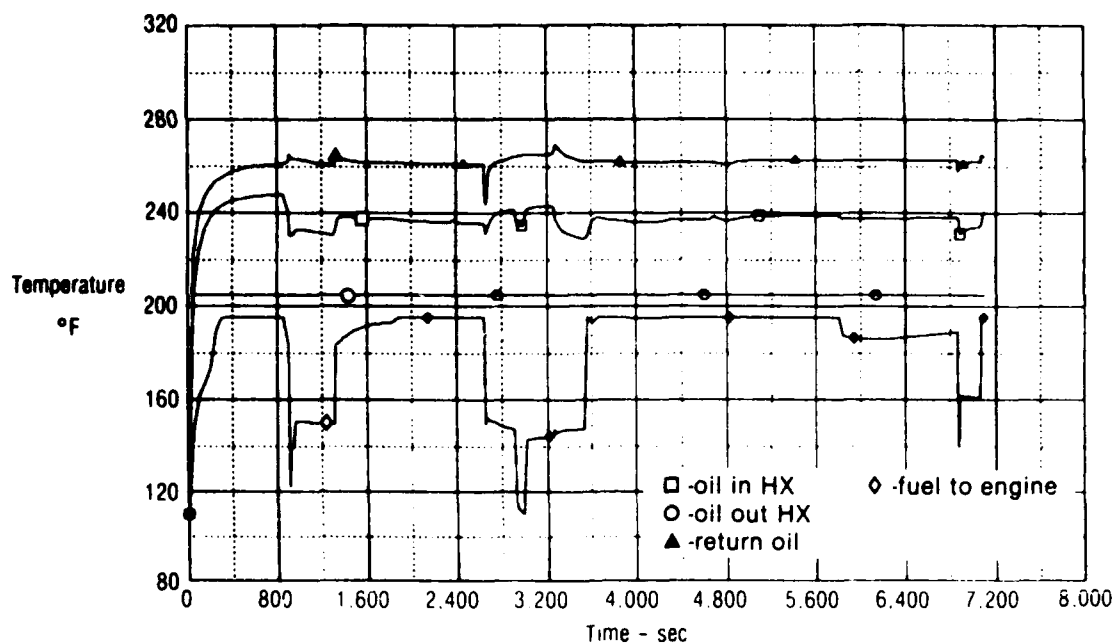


Figure 169
8,000 psi Overlap Valve
PC-1 Ram Air HX Requirements

Concept	Sys	HX	Fan	Motor	Duct	Install	Total	Total HX	ΔWeight
8,000 psi Baseline	PC-1 + PC-2	13.4	16.4	35.4	9.0	18.6	92.8	174.4	—
	Utility	13.7	14.2	29.9	7.5	16.3	81.6		
Overlap Valve	PC-1 + PC-2	7.8	11.4	24.4	6.8	12.6	63.0	121.1	-53.3
	Utility	8.4	10.0	22.7	5.4	11.6	58.1		

Fuel to oil heat exchanger weighs 9.1 lb per aircraft engine

Figure 170
Overlap Valve
Ram Air Heat Exchanger Weight Breakdown - Lb

b. Distribution System and Component Weight - Figure 171 shows the weight summary for the overlap valve concept relative to the 8,000 psi baseline. The only difference is the weight of the ram air heat exchanger, which decreased 53.3 lb. Figure 172 shows the detailed weight breakdown for the 8,000 psi overlap valve concept.

c. Life Cycle Costs - The overlap valve concept LCCs were analyzed and quantified relative to the 8,000 psi baseline configuration.

	8,000 psi		
	Baseline	Overlap Valve	ΔWeight
Flight Control Actuators	325.73	325.73	—
Engine Nozzle Actuators	284.36	284.36	—
Utility Actuators	115.40	115.40	—
Miscellaneous Components	436.25	436.25	—
Ram Air Heat Exchangers	192.60	139.30	- 53.3
Distribution System	197.92	197.92	—
Fluid - CTFE	181.77	181.77	—
Total	1,734.03	1,680.73	- 53.3

Note: Ram Air Heat Exchanger weights were established from thermal analyses includes 18.2 lb fuel/oil exchanger weight.

Figure 171
Overlap Valve Hydraulic System Weight Summary - Lb

Ram Air Heat Exchanger Requirements			
	PC-1 + PC-2 Systems	Utility System	Total
Fuel/Oil HX	9.1	9.1	18.2
Ram Air HX	7.8	8.4	16.2
Fans	11.4	10.0	21.4
Electric Motor	24.4	22.7	47.1
Duct	6.8	5.4	12.2
Installation	12.6	11.6	24.2
Total	72.1	67.2	139.3

Figure 172
Overlap Valve
Hydraulic Equipment Weight Changes - Lb

The hydraulic system components affected by the overlap valve concept are the control valves. As shown in Figure 173, no change was made in Reliability and Maintainability (R&M) relative to the baseline R&M values.

Concept	Components Affected	Type	Quantity Aircraft	Reliability MFHBF	Maintainability MTTR
Overlap Valve	No Change				

Figure 173
Overlap Valve
Reliability and Maintainability Changes

Figure 174 summarizes the LCCs for the hydraulic and aircraft systems. As shown, the overlap valve LCC decreased by \$13 M while the total aircraft LCC decreased by \$104 M. The total aircraft weight decreased 119 lb from the baseline, to 27,928 lb. The significant decrease in the LCC, resulted from the 53.3 lb decrease in the hydraulic system, as shown in Figure 171.

	8,000 psi	
	Baseline	Overlap Valve
Hydraulic System Unit Flyaway	—	- 0.024
Hydraulic System Life Cycle Cost	—	- 13.0
Aircraft System Unit Flyaway	—	- 0.088
Total Aircraft System Life Cycle Cost	—	- 104.0
Total Aircraft Weight (lb)	28,047	27,928
ΔAircraft Weight (lb)	—	- 119

Note: Life cycle costs are millions of FY dollars

Figure 174
Overlap Valve
Hydraulic and Aircraft System Life Cycle Cost/Weight Summary

3.3.3 Flow Augmentation/Load Recovery Valves - Flow augmentation/load recovery valves will reduce the system pump requirement by incorporating a jet pump on the flight control actuators to recirculate return flow to the pressure side when low resisting hinge moments occur. Reducing the system pump size will decrease pump/system heat rejection proportionally.

a. **Thermal Analysis** - The thermal analysis for the 8,000 psi CTFE flow augmentation concept was divided into two segments. The first segment was the determination of the pump heat rejection and the second was the determination of average operational and average leakage flow rates.

To determine the system pump heat rejection, an analysis was done to establish the maximum simultaneous actuator flow requirement. For this analysis, the flight control actuators incorporated the flow augmentation/load recovery valve concept. Based on testing performed at MCAIR, the flow augmentation/load recovery concept reduced the no-load actuator flow requirement 50 percent. Figure 175 shows the reduction in flow requirement for the flight control actuators.

Actuators Affected	No. of Actuators	8,000 psi No-Load Flow (gpm)	Total Flow (gpm)	8,000 psi Flow With Flow Augmentation (gpm)	Total Flow (gpm)
Aileron	2	1.08	2.16	0.54	1.08
Flaperon	2	1.31	2.62	0.65	1.31
Rudder	2	1.37	2.74	0.68	1.37
Stabilator	2	11.60	23.20	5.80	11.60
Canard	2	9.81	19.62	4.90	9.81

Figure 175
Flow Augmentation
Flow Requirements

With these flow requirements, the system pump size could be reduced by 20 percent. Further pump size reductions could be achieved if the concept was incorporated into all the utility system and engine nozzle actuators.

The pump and system total heat rejection is shown in Figure 101. As shown, the pump heat rejection equated to 42.6 horsepower per aircraft, or 10.6 horsepower per pump because of the decrease in actuator flow rates.

Figure 176 shows the thermal model parameters established for the pump. As indicated, the case drain flow decreased to 1.5 gpm and the heat rejection into the case drain and pump discharge port retained the two-thirds/one-third relationship.

Configuration	Pump Heat Rejection (hp)	H.R. Accounted for (hp)	Case Drain Flow (gpm)	H.R. to Case Drain (hp)	H.R. to Discharge (hp)	Actuator Operational Flows	Actuator Leakage Flows	System Pressure (psi)
8,000 psi Baseline	16.64	14.14 (85%)	2.0	9.47 (67%)	4.67 (33%)	Figure 122	Figure 123	8.000
Flow Augmentation	12.14	10.32 (85%)	1.5	6.91 (67%)	3.41 (33%)	Figure 177	Figure 178	8.000

Figure 176
Flow Augmentation Thermal Model Parameters

The actuator flow and leakage rates are shown in Figures 177 and 178. No change occurred relative to the 8,000 psi baseline. Figures 179 through 190 show the temperatures that were computed when the above flows were used with the thermal model mentioned in Section 3.1.2.a. These plots should be interpreted as described in Section 3.3.1.a.

Per Actuator	Actuator Flow (Unloaded)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	1.08	0.228 ¹	0.034	0.034	0.023	0.164	0.023	0.023	0.076	0.011
Flaperon	1.31	0.875 ¹	—	0.137	—	0.051	—	—	0.122	0.014
Rudder	1.37	0.175 ¹	0.043	0.015	0.208		0.015	0.015	0.143	0.143
Stabilator	11.60	1.75 ¹	1.092	0.605	0.242	1.732	0.242	0.484	1.092	0.121
Canard	9.81	1.75 ¹	0.768	0.307	0.102	1.489	0.102	0.307	0.512	0.102
Utility System										
Gun	10.88	—	—	—	—	0.107	—	—	—	—
Steering	0.90	0.64	0.64	—	—	—	—	—	—	0.64
Radar	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64
Nozzles	—	7.28	5.81	0.204	0.061	0.162	0.061	0.039	1.08	0.54
Ramps	5.91	—	0.006	0.008	0.008	0.213	0.008	0.006	0.006	—
Aileron	2.17	0.456 ¹	0.069	0.069	0.046	0.329	0.046	0.046	0.152	0.022
Flaperon	2.61	1.75 ¹	—	0.274	—	0.099	—	—	0.244	0.028

¹ = leakage

Note: Flows are mean values over flight phase in GPM

Figure 177
F-15 SMTD
Operational Flow 8,000 psi Flow Augmentation

Per Actuator	Nail Leakage (per Aircraft)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	0.130	0.130	0.092	0.092	0.104	—	0.104	0.104	0.042	0.118
Flaperon	0.50	0.50	0.50	—	0.50	0.50	0.50	0.50	—	0.45
Rudder	0.10	0.10	0.10	0.07	0.09	—	0.09	0.09	—	—
Stabilator	1.00	1.00	0.10	0.50	0.80	0.20	0.80	0.60	0.10	0.90
Canard	1.00	1.00	0.25	0.70	0.90	0.80	0.90	0.70	0.50	0.90
Utility System										
Gun	0.34	0.34	0.34	0.34	0.34	—	0.34	0.34	0.34	0.34
Steering	0.11	0.032	0.032	—	—	—	—	—	—	0.032
Radar	—	—	—	—	—	—	—	—	—	—
Nozzles	4.16	4.16	—	—	—	—	—	2.08	3.786	4.16
Ramps	1.56	1.56	1.55	1.528	1.528	1.326	1.528	1.55	1.55	1.440
Aileron	0.26	0.26	0.182	0.208	—	—	—	0.208	0.086	0.234
Flaperon	1.00	1.00	1.00	—	1.00	1.00	1.00	1.00	—	0.90

Note: Flows are mean values over flight phase in GPM

Figure 178
F-15 SMTD
Leakage Flow 8,000 psi Flow Augmentation

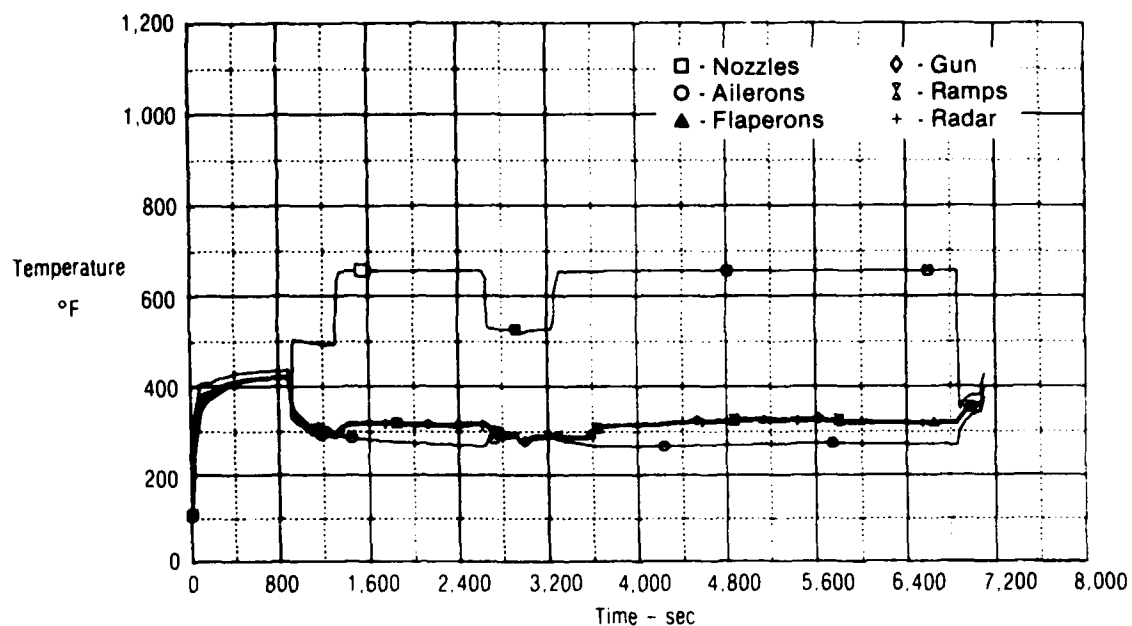


Figure 179
8,000 psi Flow Augmentation Without Ram Air HX
 Utility Actuator Exit Temperatures

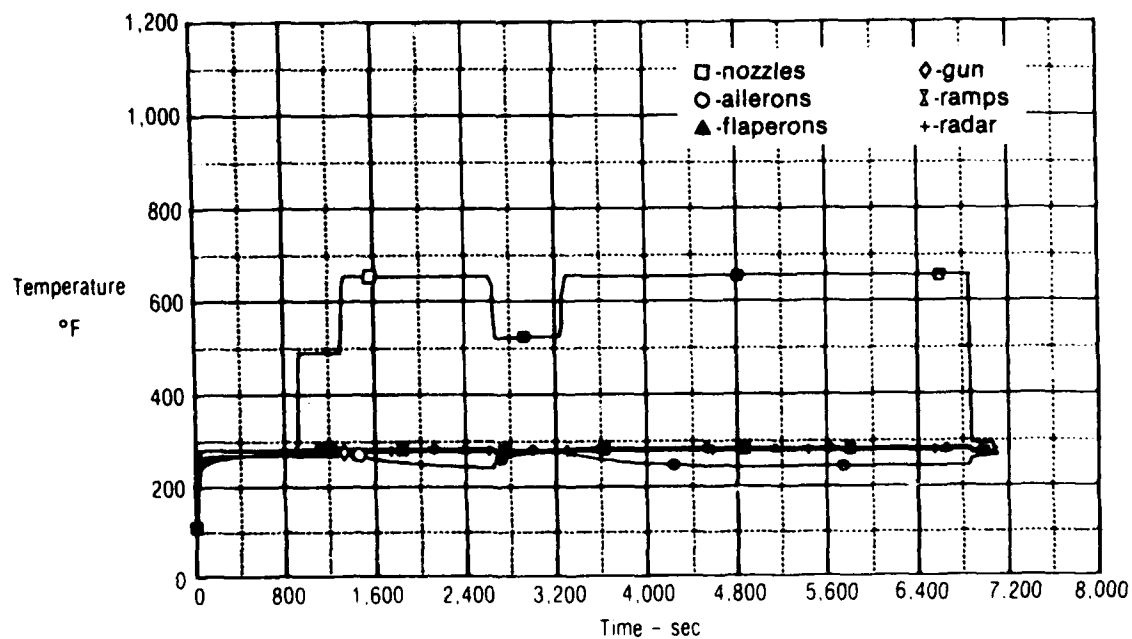


Figure 180
8,000 psi Flow Augmentation
 Utility Actuator Exit Temperatures

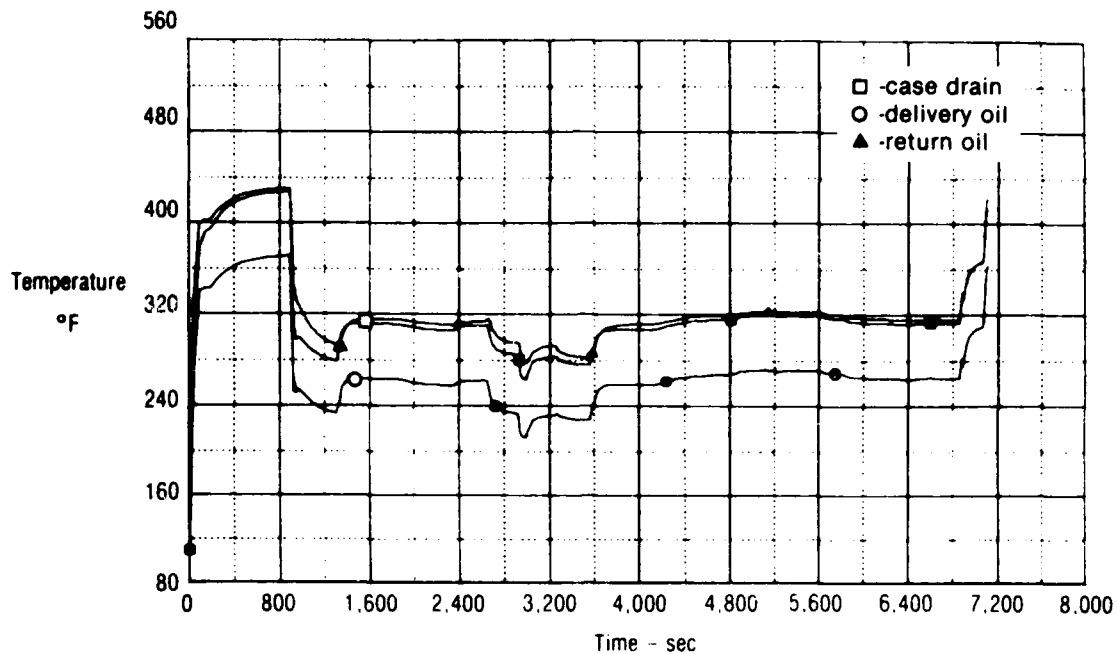


Figure 181
8,000 psi Flow Augmentation Without Ram Air HX
 Utility Pump and Return Temperatures

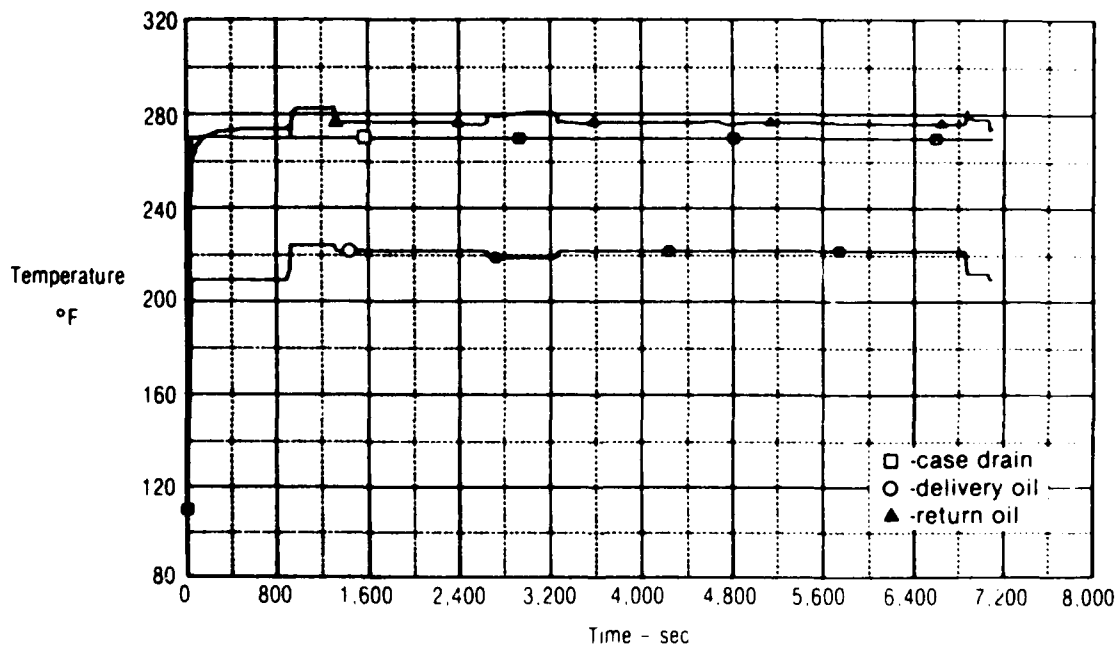


Figure 182
8,000 psi Flow Augmentation
 Utility Pump and Return Temperatures

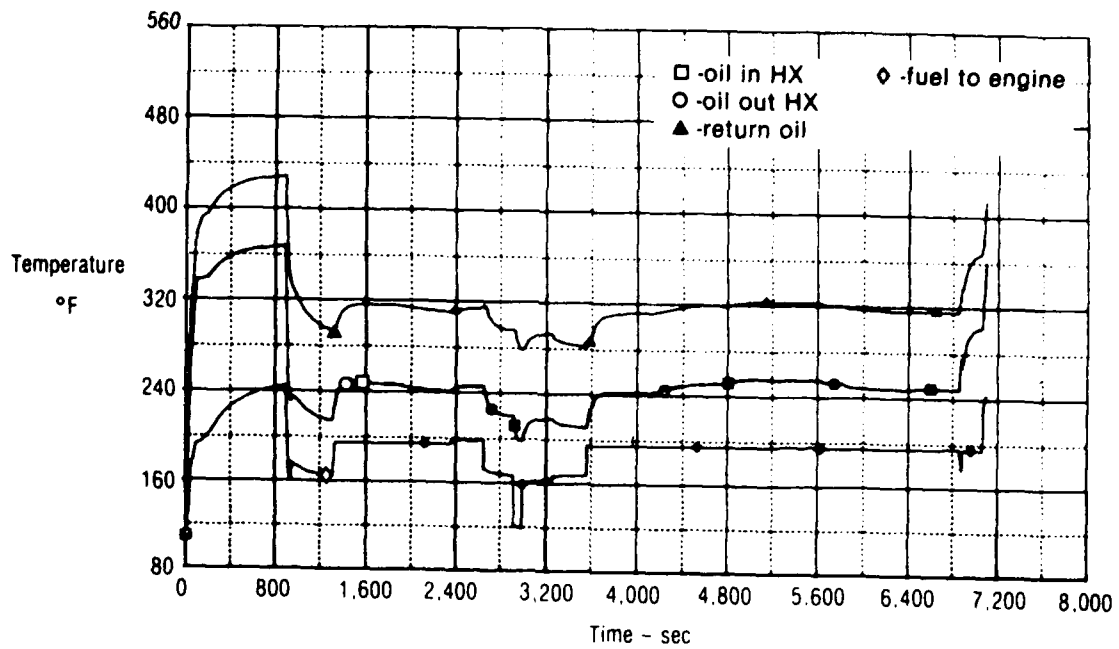


Figure 183
8,000 psi Flow Augmentation Without Ram Air HX
 Utility Ram Air HX Requirements

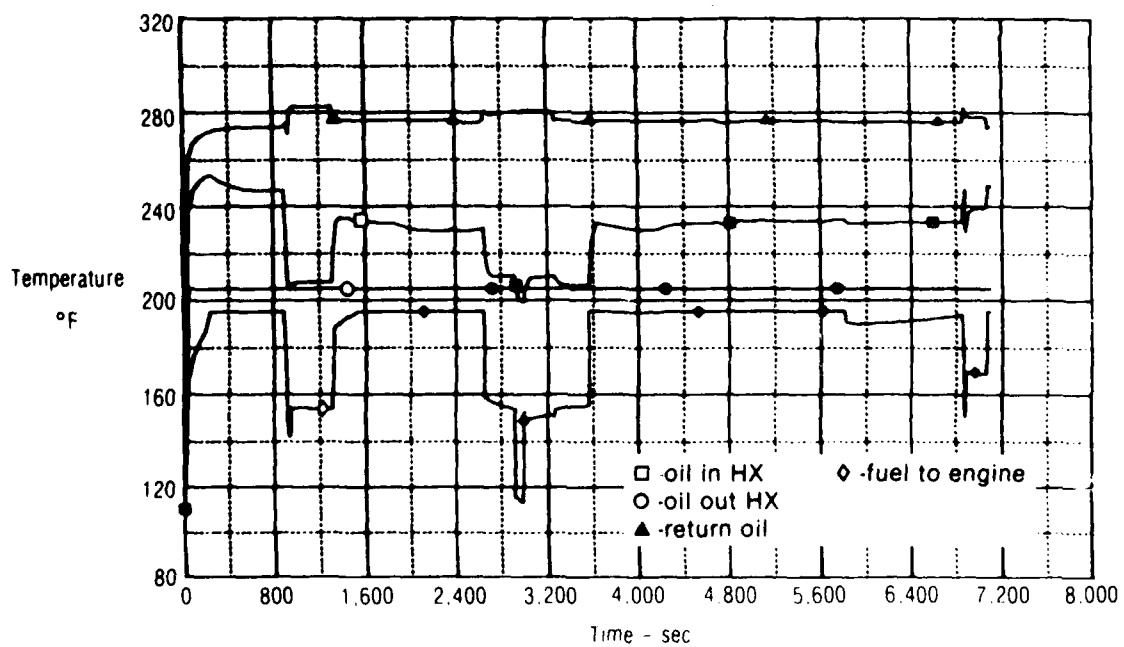


Figure 184
8,000 psi Flow Augmentation
 Utility Ram Air HX Requirements

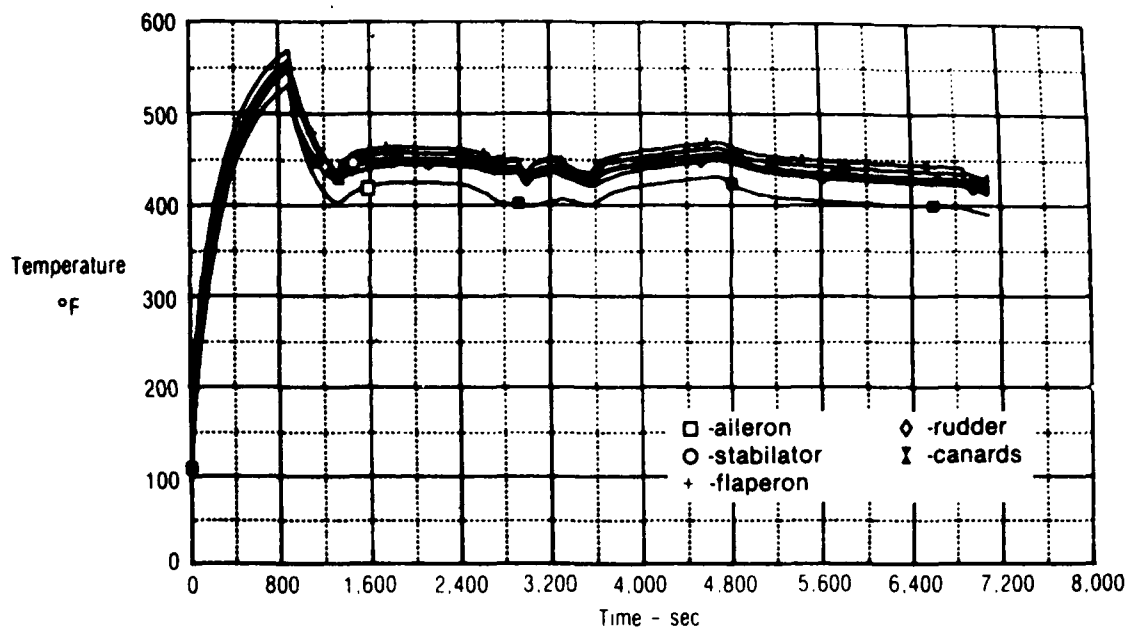


Figure 185
8,000 psi Flow Augmentation Without Ram Air HX
PC-1 Actuator Exit Temperatures

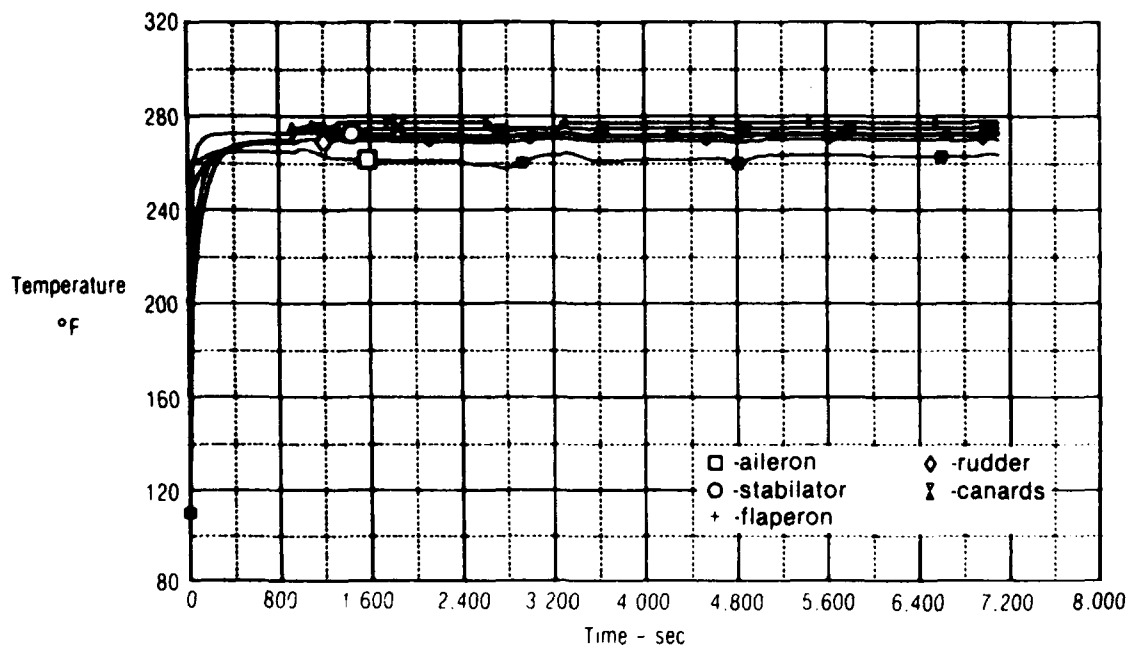


Figure 186
8,000 psi Flow Augmentation
PC-1 Actuator Exit Temperatures

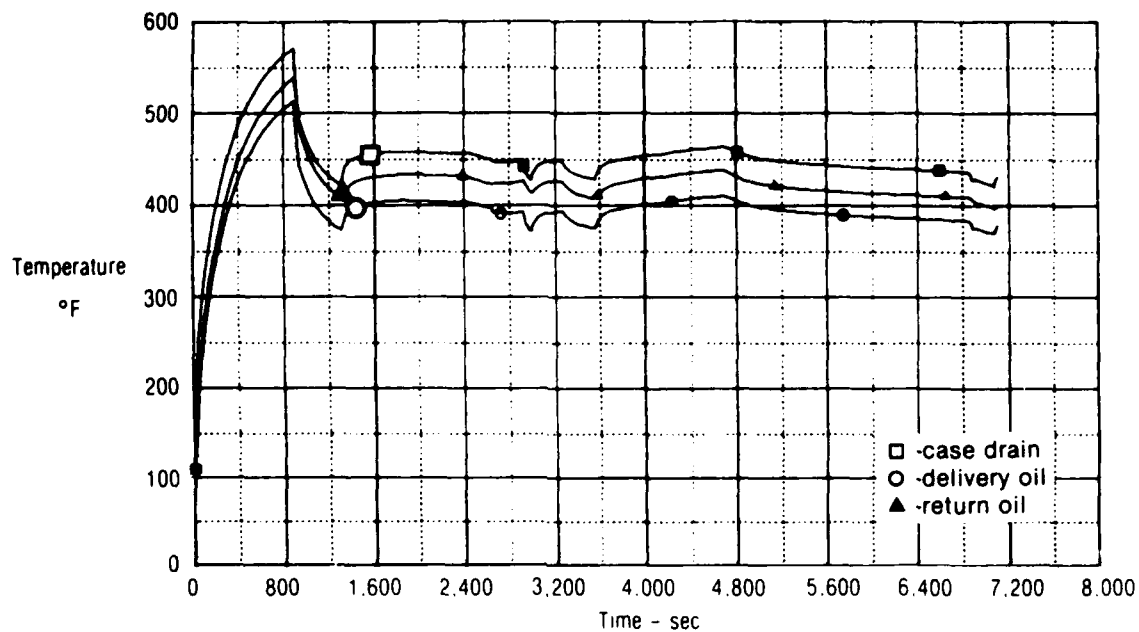


Figure 187
8,000 psi Flow Augmentation Without Ram Air HX
PC-1 Pump and Return Temperatures

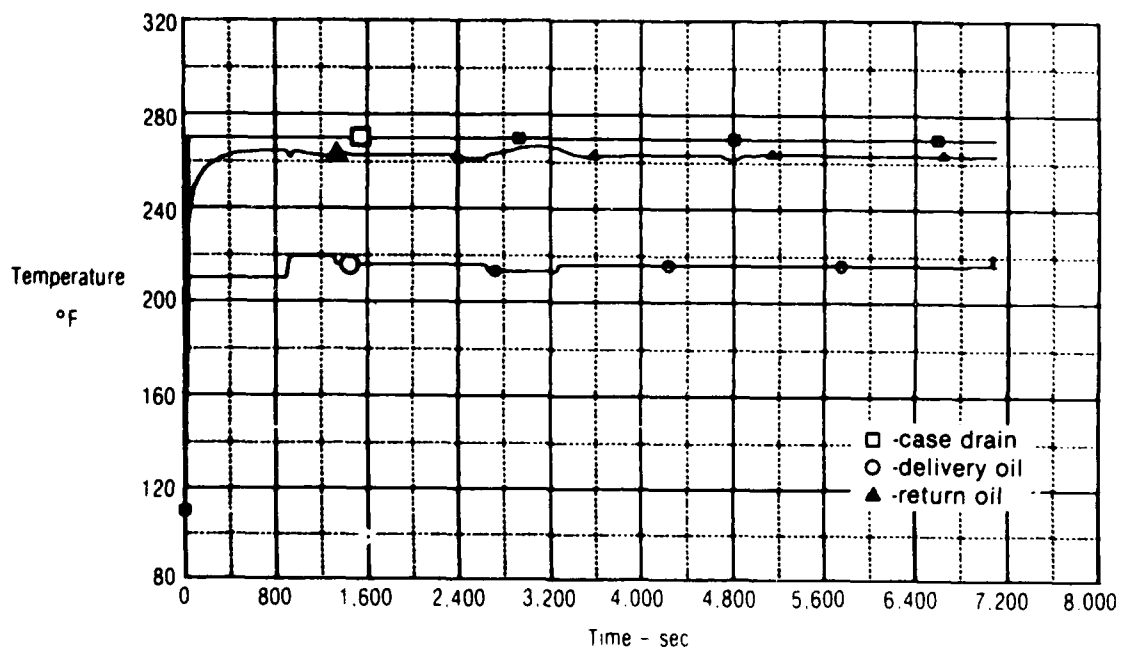


Figure 188
8,000 psi Flow Augmentation
PC-1 Pump and Return Temperatures

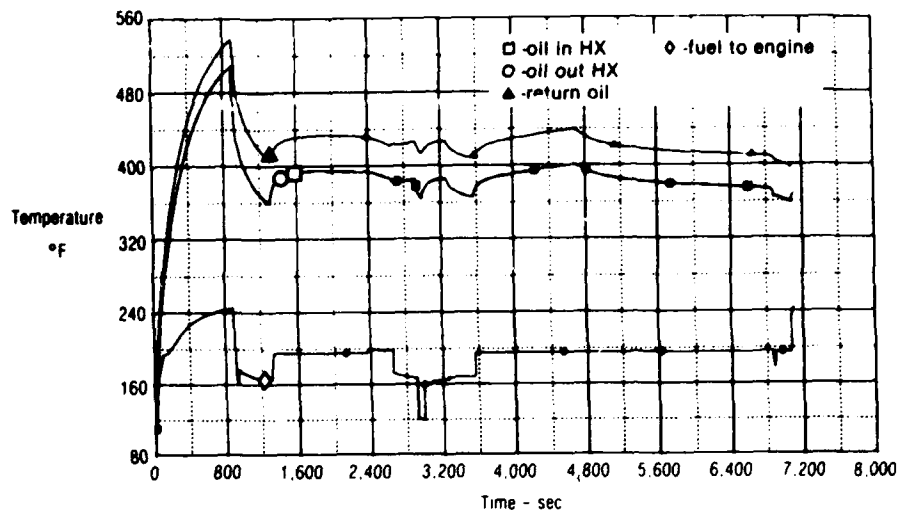


Figure 189
8,000 psi Flow Augmentation Without Ram Air HX
PC-1 Ram Air HX Requirements

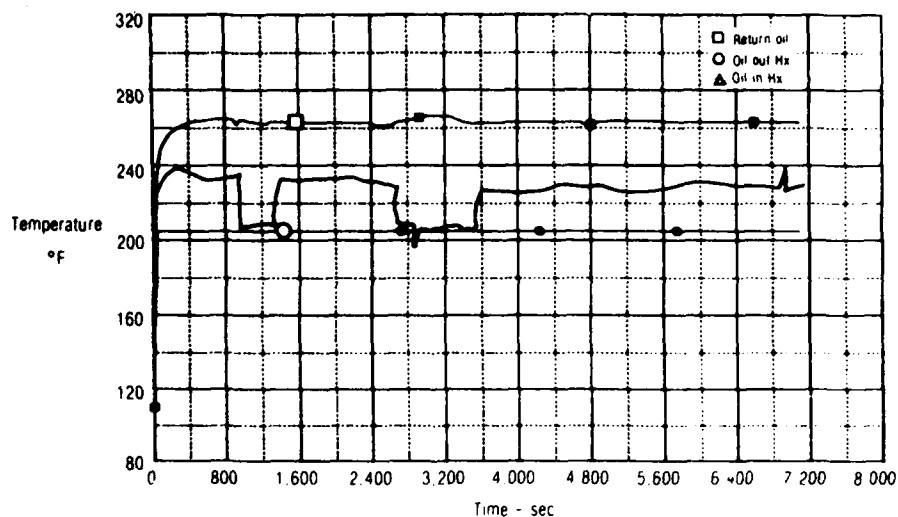


Figure 190
8,000 psi Flow Augmentation
PC-1 Ram Air HX Requirements

The weight impact of the ram air heat exchanger when utilizing the flow augmentation concept is shown in Figure 191. As shown, the total weight decreased by 10.6 lb. Note that the majority of weight was due to the fans and motors needed during ground operation.

Concept	Sys	HX	Fan	Motor	Duct	Install	Total	Total HX	ΔWeight
PC-1 8,000 psi Baseline	Utility	13.4	16.4	35.4	9.0	18.6	92.8	174.4	—
		13.7	14.2	29.9	7.5	16.3	81.6		
Flow Augmentation PC-2	Utility	12.2	15.6	32.8	8.4	17.2	86.2	163.8	- 10.6
		12.6	13.4	28.9	7.2	15.5	77.6		

Fuel to oil heat exchanger weighs 9.1 lb per aircraft engine

Figure 191
Flow Augmentation
Ram Air Heat Exchanger Weight Breakdown - Lb

b. Distribution System and Component Weight - Figure 192 shows the hydraulic weight summary for the flow augmentation/load recovery valve concept. Relative to the 8,000 psi baseline weight, the differences occurred with the flight control actuators, ram air heat exchangers, distribution system, miscellaneous components and the CTFE fluid.

	8,000 psi		
	Baseline	Flow Augmentation	ΔWeight
Flight Control Actuators	325.73	344.81	+ 19.08
Engine Nozzle Actuators	284.36	284.36	—
Utility Actuators	115.40	115.40	—
Miscellaneous Components	436.25	408.25	- 28.00
Ram Air Heat Exchangers	192.60	182.00	- 10.60
Distribution System	197.92	190.34	- 7.58
Fluid - CTFE	181.77	175.68	- 6.09
Total	1,734.03	1,700.84	- 33.19

Note: Ram Air Heat Exchanger weights were established from thermal analyses includes 18.2 lb fuel/oil exchanger weight.

Figure 192
Flow Augmentation Hydraulic System Weight Summary - Lb

The flight control actuators showed an increase from the baseline of 19.08 lb. This additional weight resulted from an analysis that quantified the dry weight of a flow augmentation device at 1.06 lb. The miscellaneous components decreased 28.00 lb because the pumps and reservoirs were smaller. The ram air heat exchanger weight decreased 10.6 lb because of the reduced system heat rejection. The distribution system weight decreased 7.58 lb because the lines going to the flight control actuators were smaller with the lower flow requirement. The fluid weight decreased 6.09 lb because of the smaller reservoir volume and the smaller distribution system.

The total hydraulic system weight decreased 33.19 lb. Figure 193 shows the detailed component weight breakdown for the 8,000 psi flow augmentation concept.

	Dry Weight (lb)	Fluid Volume (in. ³)	CTFE Fluid Weight (lb)	Wet Weight (lb)
Rudder				
Actuator	40.85	17.63	1.13	41.98
Other	3.00			3.00
Total	43.85	17.63	1.13	44.98
Flaperon				
Actuator	57.24	13.00	0.83	58.07
Total	57.24	13.00	0.83	58.07
Canards				
Actuators	94.74	89.00	5.70	100.44
Valves	9.66	7.81	0.50	10.16
Total	104.40	96.81	6.20	110.60
Ailerons				
Actuators	57.24	13.00	0.83	58.07
Valves				
Switching	9.66	7.81	0.50	10.16
Other	0.58			0.58
Total	67.48	20.81	1.33	68.81
Stabilator				
Actuators	94.74	89.00	5.70	100.44
Valves				
Switching	9.66	7.81	0.50	10.16
Other	2.08			2.08
Total	106.48	96.81	6.20	112.68
Hydraulic PC-1 and PC-2				
Valve				
Temperature Regulator	4.08	12.50	0.80	4.68
Other	0.40			0.40
Miscellaneous				
Pump	50.00	62.50	4.00	54.00
Reservoir	18.60	222.50	14.24	32.84
Other	40.44	25.00	1.60	42.04
Total	113.52	322.50	20.64	134.16
Hydraulic Utility System				
Valve				
Temperature Regulator	2.04	6.25	0.40	2.44
Other	4.50			4.50
Miscellaneous				
Pump	50.00	62.50	4.00	54.00
Reservoir	22.90	346.50	22.18	45.08
Primary Heat Exchanger	0.83	0.63	0.04	0.87
Primary HX Valve	1.17			1.17
Other	41.73	43.44	2.78	44.51
Total	123.17	459.32	29.40	152.57
Distribution System				
PC-1 and PC-2	85.08	429.44	27.48	112.56
Utility	105.26	654.89	41.91	147.17
Total	190.34	1,084.33	69.39	259.73

Ram Air Heat Exchanger Requirements			
	PC-1 + PC-2 Systems (lb)	Utility System (lb)	Total (lb)
Fuel/Oil HX	9.1	9.1	18.2
Ram Air HX	12.2	12.6	24.8
Fans	15.6	13.4	29.0
Electric Motor	32.8	28.9	61.7
Duct	8.4	7.2	15.6
Installation	17.2	15.5	32.7
Total	95.3	86.7	182.0

Figure 193
Flow Augmentation
 Hydraulic Equipment Weight Changes - Lb

c. Life Cycle Costs - The LCCs of the flow augmentation/load recovery valve concept were quantified relative to the 8,000 psi baseline configuration.

The hydraulic system components affected by the flow augmentation concept, are the flight control actuators and system pumps. As shown in Figure 194, the actuators decreased in reliability and increased in maintainability. The pump increased in reliability while maintainability remained the same as the baseline pump. These values are also based upon data received from the actuator and pump suppliers.

Concept	Components Affected	Type	Quantity Aircraft	Reliability MFHBF	Maintainability MTTR
Flow Augmentation	Canard Actuator	SWFM	2	600	6.54
	Stabilator Actuator	SWFM	2	600	6.54
	Aileron Actuator	SWFM	2	860	4.55
	Flaperon Actuator	SWFM	2	860	4.55
	Rudder Actuator	SWFM	2	1,657	6.88
	Pump	—	4	11,250	—

Note: SWFM - Simplex With Force Motor

Figure 194
Flow Augmentation
Reliability and Maintainability Changes

Figure 195 shows a summary of the hydraulics and aircraft system life cycle costs. The hydraulic system unit flyaway costs decreased by \$15,000 while the hydraulic LCCs increased by \$2 M. The unit flyaway cost decreased because of the reduced heat exchanger cost. The total hydraulic LCC increased due to higher spare parts costs resulting from heavier and more complex flight control actuators.

When the effect of the hydraulic system with flow augmentation was considered on the total aircraft system, the total LCC decreased by \$53 M.

The total aircraft weight decreased by 74 lb.

3.3.4 Dry Sump Pump - The dry sump pump concept would reduce the system heat rejection by incorporating a scavenger pump on the case drain to extract the entrained case oil. This tends to minimize the windage losses associated with having the rotating group immersed in hydraulic oil. Based on testing performed at MCAIR, pump heat rejection could be reduced 20-30 percent.

a. Thermal Analysis - The thermal analysis for the 8,000 psi CTFE dry sump pump concept was divided into two segments. The first segment was the determination of the pump heat rejection and the other was the determination of average operational and average leakage flow rates.

	8,000 psi	
	Baseline	Flow Augmentation
Hydraulic System Unit Flyaway	—	- 0.015
Hydraulic System Life Cycle Cost	—	+ 2.0
Aircraft System Unit Flyaway	—	- 0.057
Total Aircraft System Life Cycle Cost	—	- 53.0
Total Aircraft Weight (lb)	28,047	27,973
ΔAircraft Weight (lb)	—	- 74

Note: Life cycle costs are millions of FY dollars

Figure 195
Flow Augmentation
 Hydraulic and Aircraft System Life Cycle Cost/Weight Summary

The system pump heat rejection is shown in Figure 101 for the dry sump pump design. As shown, the total heat rejection is 42.4 horsepower per aircraft, or 10.6 horsepower per pump. This pump heat rejection is 25 percent less than the baseline configuration.

Figure 196 shows the thermal model parameters established for the pump. The case drain flow remained at 2 gpm and the heat rejection distribution to the case drain and pump discharge port remained the same; two-thirds/one-third relationship as the baseline.

Configuration	Pump Heat Rejection (hp)	H.R. Accounted for (hp)	Case Drain Flow (gpm)	H.R. to Case Drain (hp)	H.R. to Discharge (hp)	Actuator Operational Flows	Actuator Leakage Flows	System Pressure (psi)
8,000 psi Baseline	16.64	14.14 (85%)	2.0	9.47 (67%)	4.67 (33%)	Figure 122	Figure 123	8,000
Dry Sump Pump	12.48	10.60 (85%)	2.0	7.10 (67%)	3.50 (33%)	Figure 197	Figure 198	8,000

Figure 196
 Dry Sump Pump Thermal Model Parameters

The actuator flow and leakage rates are shown in Figures 197 and 198. No change occurred relative to the 8,000 psi baseline.

Figures 199 through 210 show the temperatures computed when the above data was used with the thermal computer model mentioned in Section 3.1.2.a. These plots should be interpreted as described in Section 3.3.1.a.

Per Actuator	Actuator Flow (Unloaded)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	1.08	0.228 ¹	0.034	0.034	0.023	0.164	0.023	0.023	0.076	0.011
Flaperon	1.31	0.875 ¹	—	0.137	—	0.051	—	—	0.122	0.014
Rudder	1.37	0.175 ¹	0.043	0.015	0.015	0.208	0.015	0.015	0.143	0.143
Stabilator	11.60	1.75 ¹	1.092	0.605	0.242	1.732	0.242	0.484	1.092	0.121
Canard	9.81	11.75 ¹	0.768	0.307	0.102	1.489	0.102	0.307	0.512	0.102
Utility System										
Gun	10.88	—	—	—	—	0.107	—	—	—	—
Steering	0.90	0.64	0.64	—	—	—	—	—	—	0.64
Radar	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64
Nozzles	—	7.28	5.81	0.204	0.061	0.162	0.061	0.039	1.08	0.54
Ramps	5.91	—	0.006	0.008	0.008	0.213	0.008	0.006	0.006	—
Aileron	2.17	0.456 ¹	0.069	0.069	0.046	0.329	0.046	0.046	0.152	0.022
Flaperon	2.61	1.75 ¹	—	0.274	—	0.099	—	—	0.244	0.028

¹ = leakage

Note: Flows are mean values over flight phase in GPM

Figure 197
F-15 SMTD
 Operational Flow 8,000 psi Dry Sump Pump

Per Actuator	Null Leakage (per Aircraft)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	0.130	0.130	0.092	0.092	0.104	—	0.104	0.104	0.042	0.118
Flaperon	0.50	0.50	0.50	—	0.50	0.50	0.50	0.50	—	0.45
Rudder	0.10	0.10	0.10	0.07	0.09	—	0.09	0.09	—	—
Stabilator	1.00	1.00	0.10	0.50	0.80	0.20	0.80	0.60	0.10	0.90
Canard	1.00	1.00	0.25	0.70	0.90	0.80	0.90	0.70	0.50	0.90
Utility System										
Gun	0.34	0.34	0.34	0.34	0.34	—	0.34	0.34	0.34	0.34
Steering	0.11	0.032	0.032	—	—	—	—	—	—	0.032
Radar	—	—	—	—	—	—	—	—	—	—
Nozzles	4.16	4.16	—	—	—	—	—	2.08	3.786	4.16
Ramps	1.56	1.56	1.55	1.528	1.528	1.326	1.528	1.55	1.55	1.440
Aileron	0.26	0.26	0.182	0.208	—	—	—	0.208	0.086	0.234
Flaperon	1.00	1.00	1.00	—	1.00	1.00	1.00	1.00	—	0.90

Note: Flows are mean values over flight phase in GPM

Figure 198
F-15 SMTD
 Leakage Flow 8,000 psi Dry Sump Pump

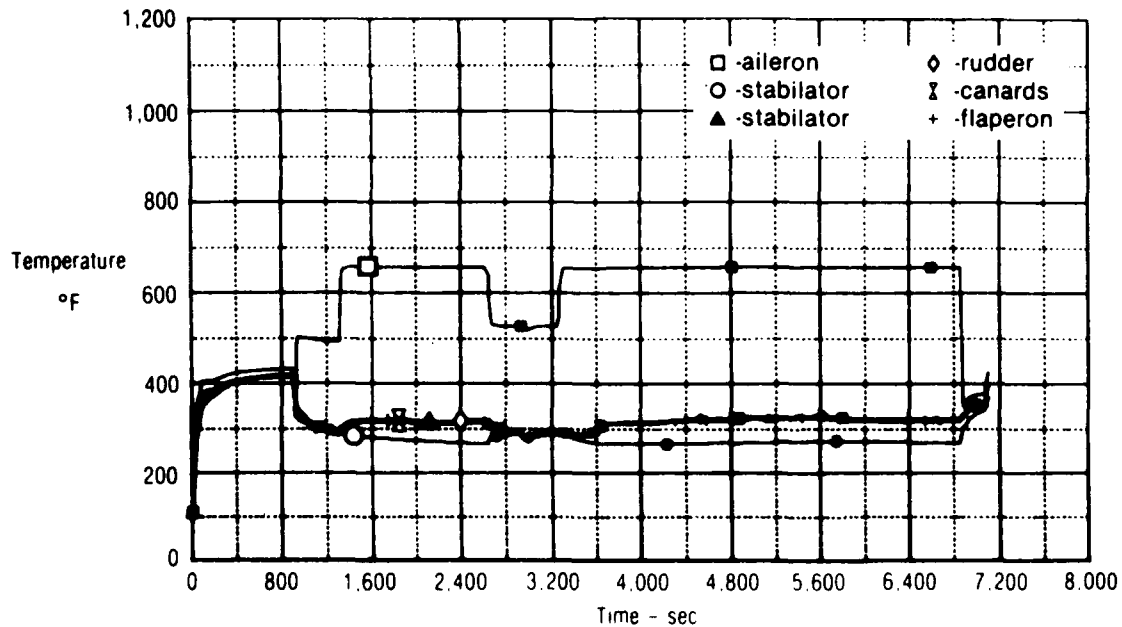


Figure 199
 8,000 psi Dry Sump Pump Without Ram Air HX
 Utility Actuator Exit Temperatures

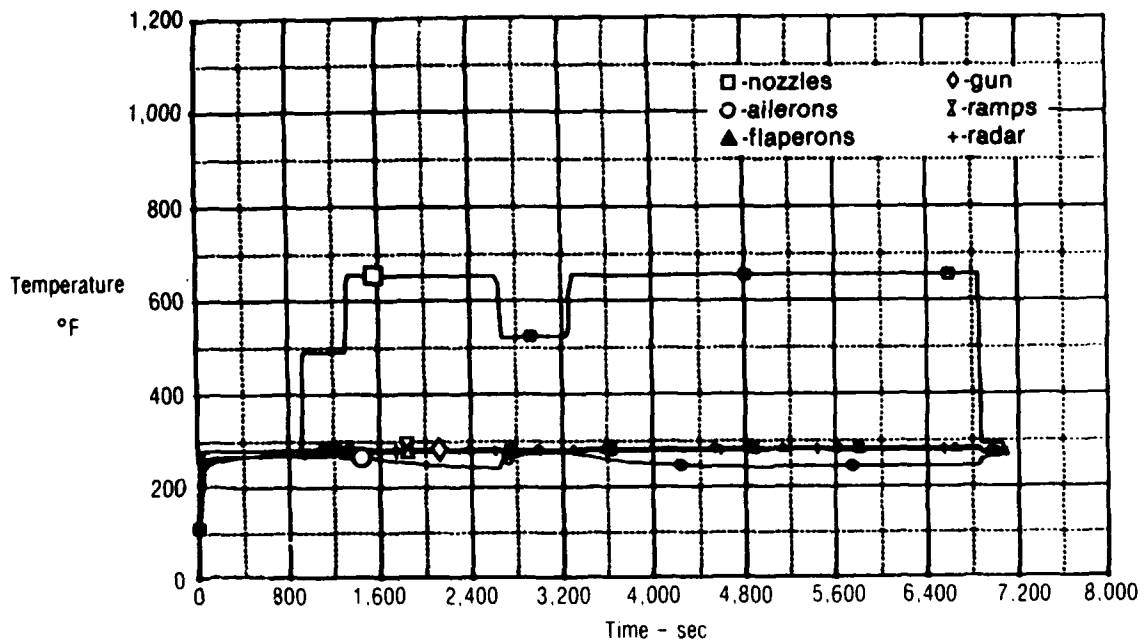


Figure 200
8,000 psi Dry Sump Pump
Utility Actuator Exit Temperatures

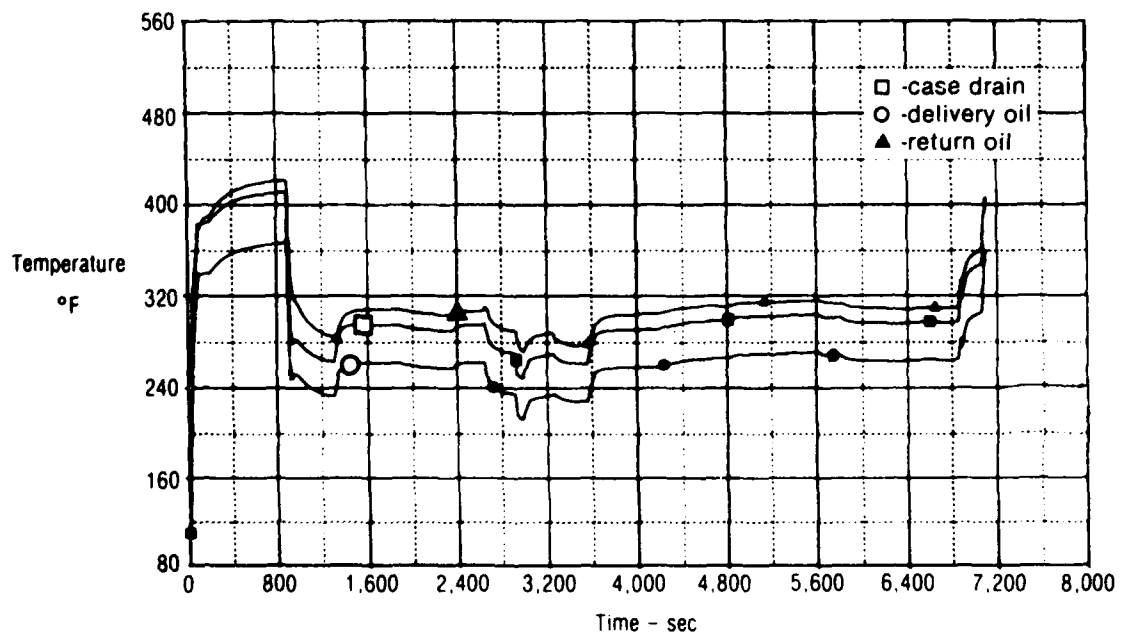


Figure 201
8,000 psi Dry Sump Pump Without Ram Air HX
Utility Pump and Return Temperatures

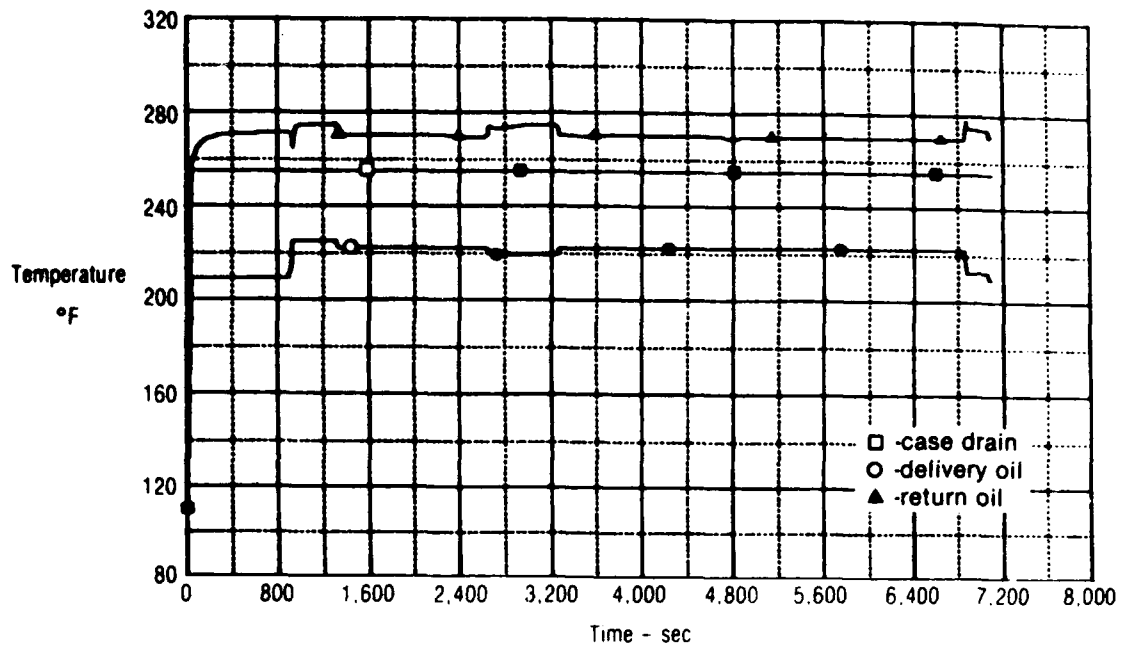


Figure 202
8,000 psi Dry Sump Pump
 Utility Pump and Return Temperatures

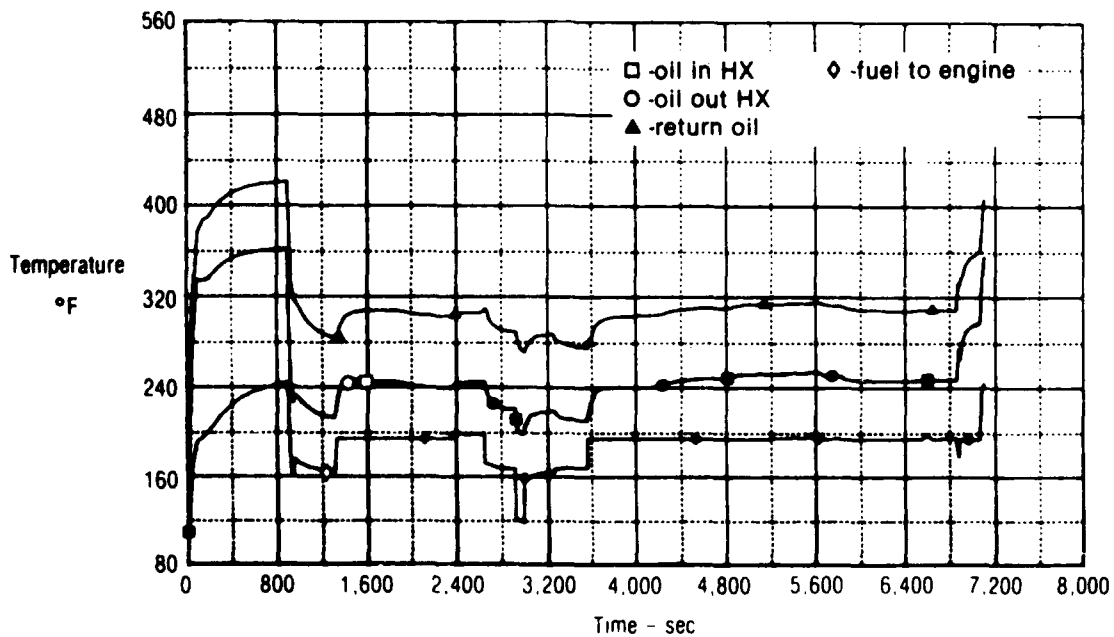


Figure 203
8,000 psi Dry Sump Pump Without Ram Air HX
 Utility Ram Air HX Requirements

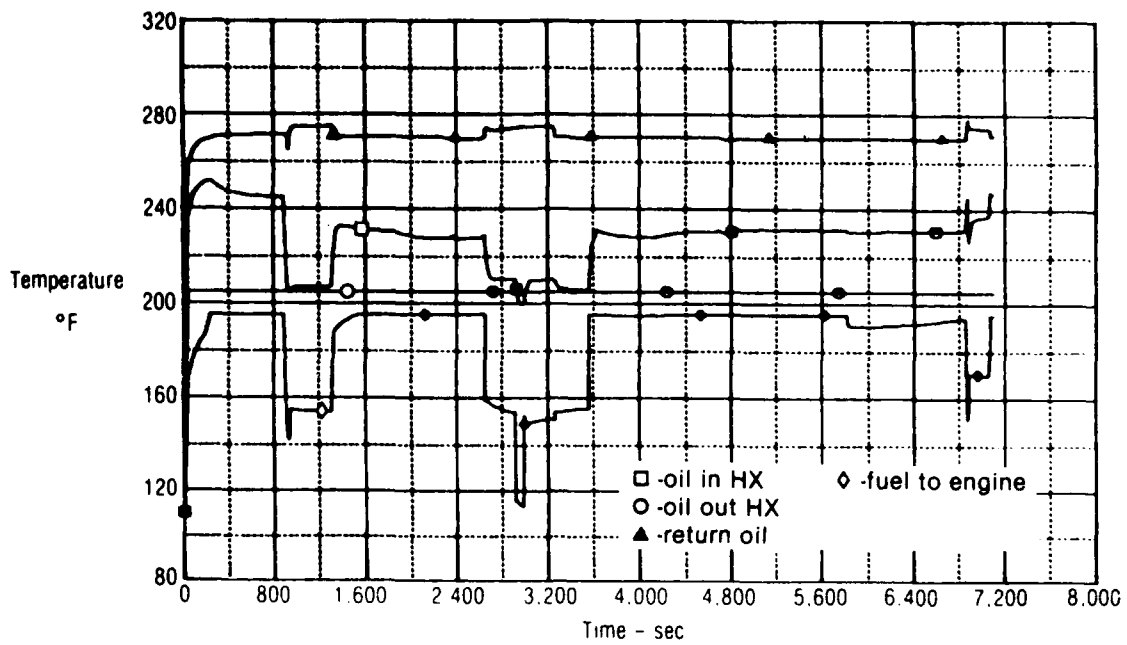


Figure 204
8,000 psi Dry Sump Pump
 Utility Ram Air HX Requirements

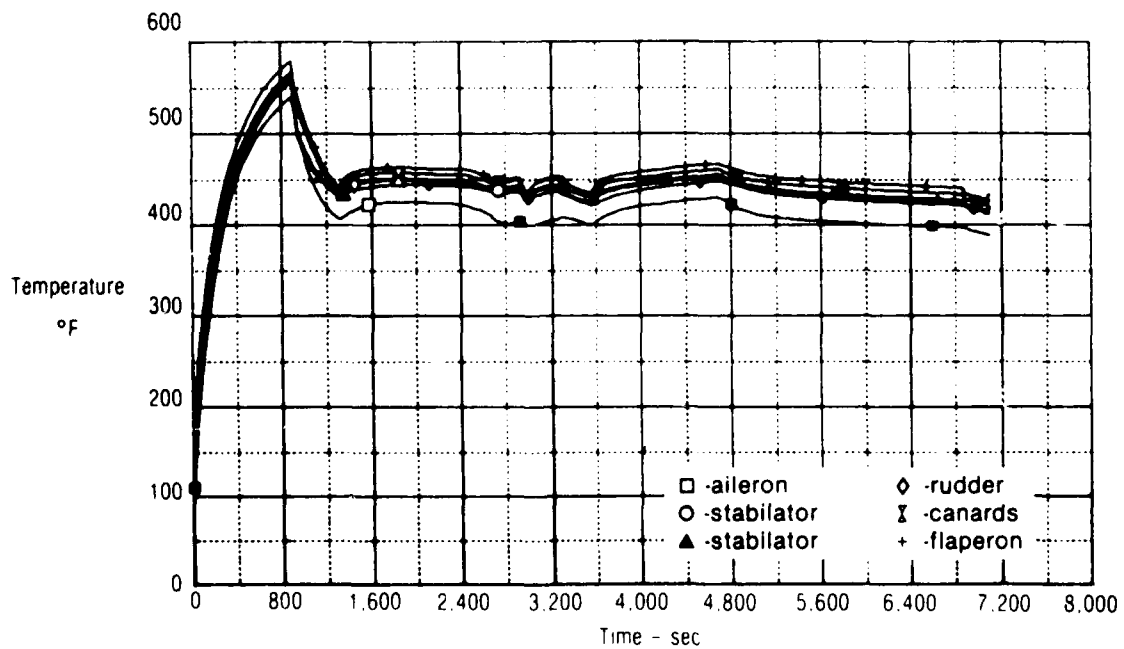


Figure 205
8,000 psi Dry Sump Pump Without Ram Air HX
 PC-1 Actuator Exit Temperatures

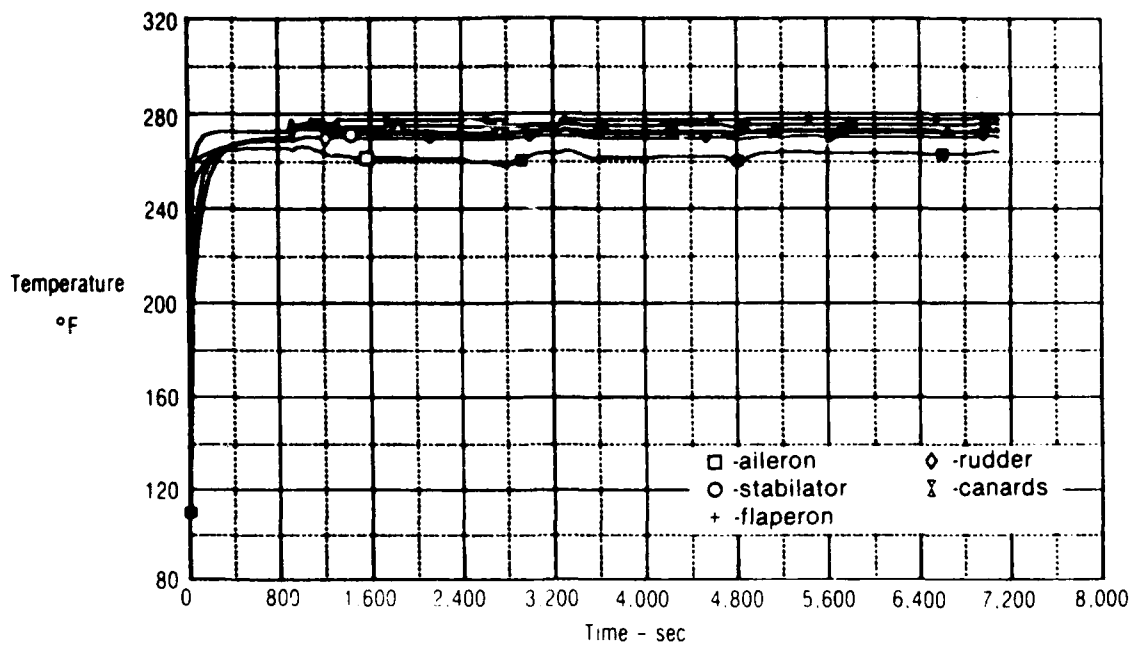


Figure 206
8,000 psi Dry Sump Pump
PC-1 Actuator Exit Temperatures

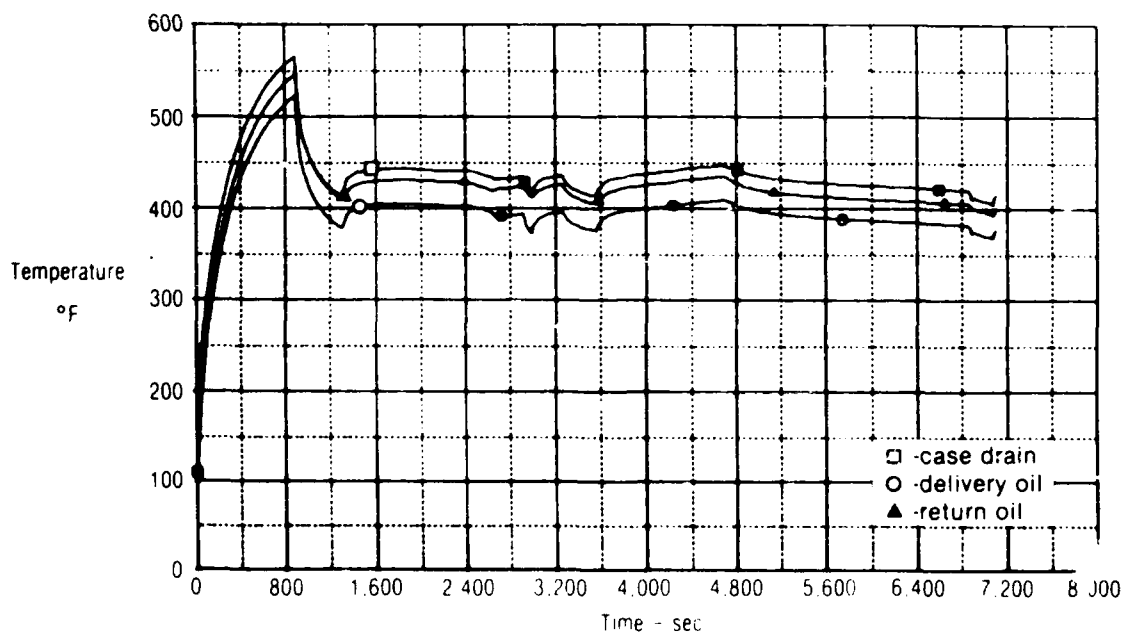


Figure 207
8,000 psi Dry Sump Pump Without Ram Air HX
PC-1 Pump and Return Temperatures

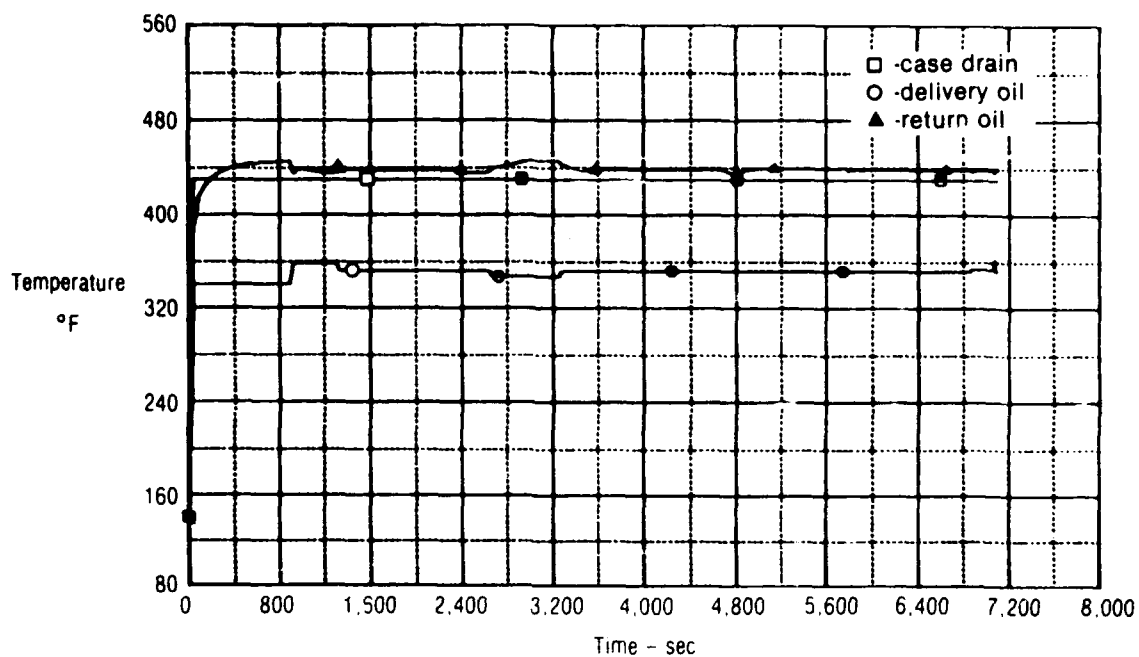


Figure 208
8,000 psi Dry Sump Pump
PC-1 Pump and Return Temperatures

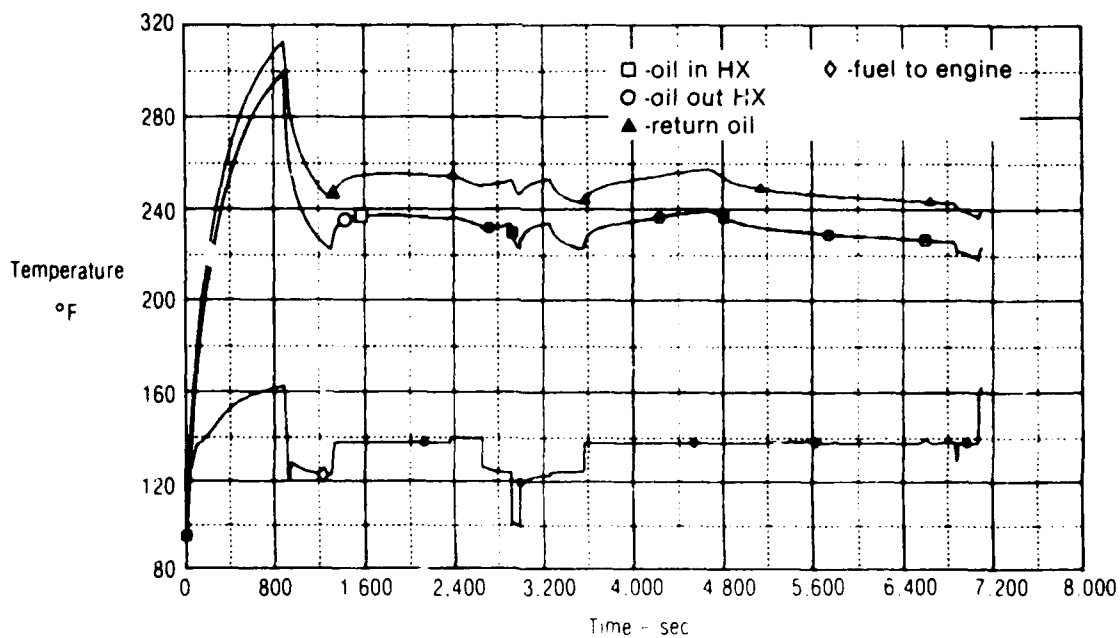


Figure 209
8,000 psi Dry Sump Pump Without Ram Air HX
PC-1 Ram Air HX Requirements

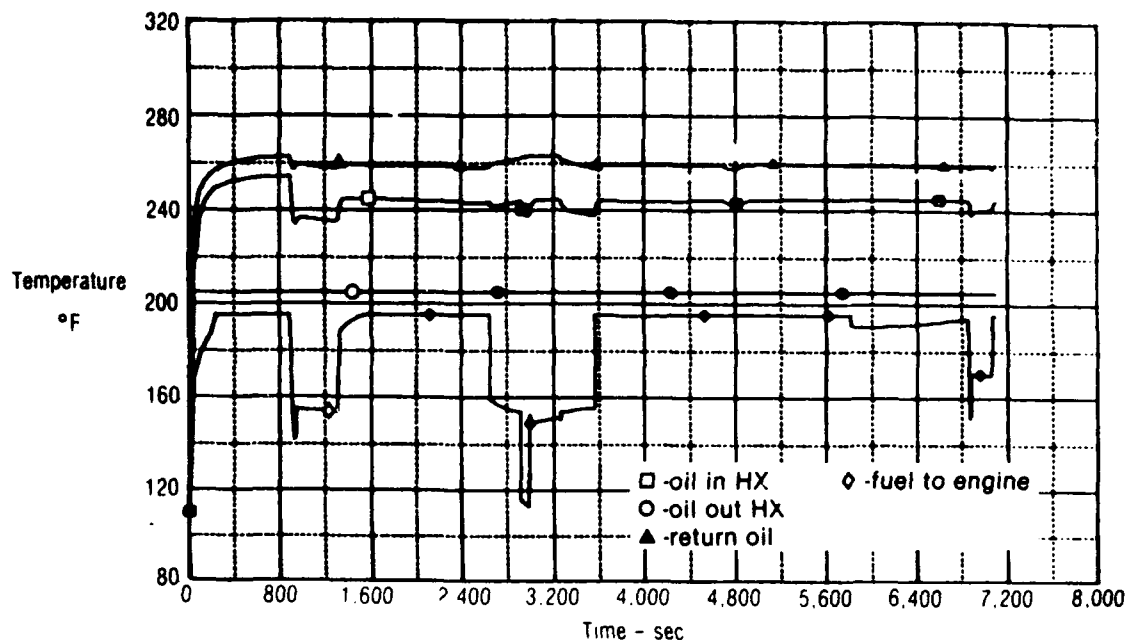


Figure 210
8,000 psi Dry Sump Pump
PC-1 Ram Air HX Requirements

The ram air heat exchanger weight impact due to the dry sump pump concept is shown in Figure 211. The analysis and design technique for the ram air heat exchangers is described in Section 3.3.1.a. As shown, the total weight decreased by 7.9 lb. Note that the majority of weight was due to the fans and motors needed during ground operation.

Concept	Sys	HX	Fan	Motor	Duct	Install	Total	Total HX	ΔWeight
PC-1	Utility	13.4	16.4	35.4	9.0	18.6	92.8	174.4	—
8,000 psi Baseline		13.7	14.2	29.9	7.5	16.3	81.6		
Dry Sump Pump	Utility	12.2	15.6	34.2	8.6	17.6	88.2	166.5	- 7.9
PC-2		12.7	13.6	29.0	7.3	15.7	78.3		

Fuel to oil heat exchanger weighs 9.1 lb per aircraft engine

Figure 211
Dry Sump Pump
Ram Air Heat Exchanger Weight Breakdown - Lb

b. Distribution System and Component Weight - Figure 212 shows the hydraulic weight summary for the dry sump pump concept. Relative to the 8,000 psi baseline weight, the differences occurred with the miscellaneous components, ram air heat exchangers and the CTFE fluid.

	8,000 psi		
	Baseline	Dry Sump	ΔWeight
Flight Control Actuators	325.73	325.73	—
Engine Nozzle Actuators	284.36	284.36	—
Utility Actuators	115.40	115.40	—
Miscellaneous Components	436.25	436.25	+ 7.54
Ram Air Heat Exchangers	192.60	184.70	- 7.90
Distribution System	197.92	197.92	—
Fluid - CTFE	181.77	166.41	- 15.36
Total	1,734.03	1,718.31	- 15.72

Note: Ram Air Heat Exchange weights were established from thermal analyses includes 18.2 lb fuel/air exchanger weight

Figure 212
Dry Sump Pump Hydraulic System Weight Summary - Lb

The miscellaneous components weight increased 7.54 lb. This increase was due to the addition of a scavenger pump to each system pump, and was slightly offset because the reservoirs decreased in weight. The ram air heat exchanger weight decreased 7.9 lb. because of the decreased system heat rejection. The fluid weight decreased 15.36 lb. because the evacuated pump case oil was returned to the reservoir. This volume of oil was subtracted from the total required system volume. The total hydraulic system weight decreased 15.72 lb. Figure 213 shows the detailed component weight breakdown for the 8,000 psi dry sump pump concept.

c. Life Cycle Costs - The dry sump pump concept LCC was analyzed and quantified relative to the 8,000 psi baseline configuration.

The system component affected by the dry sump pump concept is the hydraulic pump, as shown in Figure 214. Based on inputs from pump suppliers, the pump reliability decreased and maintainability increased slightly.

	Dry Weight (lb)	Fluid Volume (in. ³)	Fluid Weight (lb) CTFE	Wet Weight (lb)
Hydraulic PC-1 and PC-2				
Valve				
Temperature Regulator	4.08	12.50	0.80	4.68
Other	0.40			0.40
Miscellaneous				
Pump	70.00	20.00	1.28	71.28
Reservoir	16.11	164.00	10.50	26.61
Other	40.44	25.00	1.60	42.04
Total	131.03	221.50	14.18	145.21
Hydraulic Utility System				
Valve				
Temperature Regulator	2.04	6.25	0.40	2.44
Other	4.50			4.50
Miscellaneous				
Pump	70.00	20.00	1.28	71.28
Reservoir	20.93	288.00	18.43	39.36
Primary Heat Exchanger	0.83	0.63	0.04	0.87
Primary HX Valve	1.17			1.17
Other	41.73	43.44	2.78	44.51
Total	141.20	358.32	22.93	164.13

Ram Air Heat Exchanger Requirements			
	PC-1 + PC-2 Systems (lb)	Utility System (lb)	Total (lb)
Fuel/Oil HX	9.1	9.1	18.2
Ram Air HX	12.2	12.7	24.9
Fans	15.6	13.6	29.2
Electric Motor	34.2	29.0	63.2
Duct	8.6	7.3	15.9
Installation	17.6	15.7	33.3
Total	97.3	87.4	184.7

Figure 213
Dry Sump Pump
Hydraulic Equipment Weight Changes - Lb

Concept	Components Affected	Type	Quantity Aircraft	Reliability MFHBF	Maintainability MTTR
Dry Sump Pump	Pump	—	4	8,041	5.90

Figure 214
Dry Sump Pump
Reliability and Maintainability Changes

Figure 215 compares the hydraulic and aircraft system LCC to the 8,000 psi CTFE baseline. As shown, the hydraulic system unit flyaway costs increased \$23,000 and the LCC increased \$22 M. This was due to the added complexity and cost of the dry sump pump. When the hydraulic system was analyzed with the total aircraft system, the total LCC decreased \$4 M from the baseline. This cost decrease was due to the reduced hydraulic system weight. The total aircraft weight decreased 35 lb.

	8,000 psi	
	Baseline	Dry Sump
Hydraulic System Unit Flyaway	—	+ 0.023
Hydraulic System Life Cycle Cost	—	+ 22.0
Aircraft System Unit Flyaway	—	+ 0.006
Total Aircraft System Life Cycle Cost	—	- 4.0
Total Aircraft Weight (lb)	28,047	28,012
ΔAircraft Weight (lb)	—	- 35

Notes:

(1) Life cycle costs are millions of FY dollars

(2) Total aircraft equipment cost is based on 500 shipsets

Figure 215
Dry Sump Pump
Hydraulic and Aircraft System Life Cycle Cost/Weight Summary

3.3.5 Intelligent Pump - The intelligent pump, shown in Figure 105, will reduce system heat rejection by producing only enough pressure to achieve commanded actuator positions. Studies of F-18 Yuma Arizona test data, indicated the aircraft could function at one-third of full system operating pressure for 93 percent of a mission, and full system operating pressure for only 7 percent. The exception was the combat phase, where it increased to 50 percent/50 percent. The operation at lower pressure yielded a significant savings in heat rejection.

a. Thermal Analysis - The thermal analysis for the intelligent pump concept was divided into two segments. The first segment was the determination of the pump heat rejection and the other was the determination of system average operational and average leakage flow rates.

The intelligent pump heat rejection is shown in Figure 101. There are two levels of heat rejection, 3,000 and 8,000 psi. The 8,000 psi total heat rejection was the same as the baseline configuration. A lower heat rejection occurred when the pump output pressure was decreased to 3,000 psi.

Figure 216 shows the thermal parameters established for the pump. As indicated, the heat rejection decreased 50 percent with 3,000 psi and the case drain flow decreased to 1.0 gpm at 3,000 psi. The heat rejection to the case drain and pump discharge port remained at the two-thirds/one-third relationship for each condition.

Configuration	Pump Heat Rejection (hp)	H.R. Accounted for (hp)	Case Drain Flow (gpm)	H.R. to Case Drain (hp)	H.R. to Discharge (hp)	Actuator Operational Flows	Actuator Leakage Flows	System Pressure (psi)
8,000 psi Baseline	16.64	14.14 (85%)	2.0	9.47 (67%)	4.67 (33%)	Figure 122	Figure 123	8,000
Intelligent Pump	16.64/8.32 (8,000/3,000)	14.14/7.07 (85%)	2.0/1.0	9.47/4.74 (67%)	4.66/2.33 (33%)	Figure 217	Figure 218	8,000/3,000

Figure 216
Intelligent Pump Thermal Model Parameters

The actuator flows and leakages are shown in Figures 217 through 220. The flow rates are tabulated for 3,000 and 8,000 psi system operation. As indicated, the 8,000 psi flow and leakage values are the same as the baseline. The actuator areas were sized to meet the velocity requirement with 8,000 psi. Because the velocity was reduced when operating at 3,000 psi, less flow was necessary to maintain that rate. Based on the Yuma, Arizona test data, the leakage/operational flows and heat rejection were assumed to be 93 percent/7 percent during all mission segments except 50 percent/50 percent during combat.

Figures 221 through 232 show the temperatures computed when the above flows and pump heat rejections were used with the thermal computer model, as mentioned in Section 3.1.2.a. These plots should be interpreted as described in Section 3.3.1.a.

The weight impact of the ram air heat exchanger when utilizing the intelligent pump concept, is shown in Figure 233. The analysis and design technique for the ram air heat exchanger is described in Section 3.3.1.a. The heat exchanger with the intelligent pump, as indicated in Figure 148, was sized by the combat flight phase, because of the increased heat load at 8,000 psi. The fans were sized for ground taxi conditions. As shown, the total weight decreased 115 lb. The low level of heat rejection experienced with this concept, required only a small increase in heat exchanger capacity.

Per Actuator	Actuator Flow (Unloaded)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	1.08	0.228 ¹	0.034	0.034	0.023	0.164	0.023	0.023	0.076	0.011
Flaperon	1.31	0.875 ¹	—	0.137	—	0.051	—	—	0.122	0.014
Rudder	1.37	0.175 ¹	0.043	0.015	0.015	0.208	0.015	0.015	0.143	0.143
Stabilator	11.60	1.75 ¹	1.092	0.605	0.242	1.732	0.242	0.484	1.092	0.121
Canard	9.81	1.75 ¹	0.768	0.307	0.102	1.489	0.102	0.307	0.512	0.102
Utility System										
Gun	10.88	—	—	—	—	0.107	—	—	—	—
Steering	0.90	0.64	0.64	—	—	—	—	—	—	0.64
Radar	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64
Nozzles	—	7.28	5.81	0.204	0.061	0.162	0.061	0.039	1.08	0.54
Ramps	5.91	—	0.006	0.008	0.008	0.213	0.008	0.006	0.006	—
Aileron	2.17	0.456 ¹	0.069	0.069	0.046	0.329	0.046	0.046	0.152	0.022
Flaperon	2.61	1.75 ¹	—	0.274	—	0.099	—	—	0.244	0.028

¹ = leakage

Note: Flows are mean values over flight phase in GPM

Figure 217
F-15 SMTD
 Operational Flow 8,000 psi Intelligent Pump

Per Actuator	Null Leakage (Per Aircraft)	Flight Phase								
		Taxi	STOL	Chmb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	0.130	0.130	0.092	0.092	0.104	—	0.104	0.104	0.042	0.118
Flaperon	0.50	0.50	0.50	—	0.50	0.50	0.50	0.50	—	0.45
Rudder	0.10	0.10	0.10	0.07	0.09	—	0.09	0.09	—	—
Stabilator	1.00	1.00	0.10	0.50	0.80	0.20	0.80	0.60	0.10	0.90
Canard	1.00	1.00	0.25	0.70	0.90	0.80	0.90	0.70	0.50	0.90
Utility System										
Gun	0.34	0.34	0.34	0.34	0.34	—	0.34	0.34	0.34	0.34
Steering	0.11	0.032	0.032	—	—	—	—	—	—	0.032
Radar	—	—	—	—	—	—	—	—	—	—
Nozzles	4.16	4.16	—	—	—	—	—	2.08	3.786	4.16
Ramps	1.56	1.56	1.55	1.528	1.528	1.326	1.528	1.55	1.55	1.440
Aileron	0.26	0.26	0.182	0.208	—	—	—	0.208	0.086	0.234
Flaperon	1.00	1.00	1.00	—	1.00	1.00	1.00	1.00	—	0.90

Note: Flows are mean values over flight phase in GPM

Figure 218
F-15 SMTD
Leakage Flow 8,000 psi Intelligent Pump

Per Actuator	Actuator Flow (Unloaded)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	—	0.086	0.013	0.013	0.013	0.009	0.009	0.009	0.029	0.004
Flaperon	—	0.328	—	0.051	—	0.019	—	—	0.046	0.005
Rudder	—	0.066	0.016	0.006	0.006	0.078	0.006	0.006	0.054	0.054
Stabilator	—	0.656	0.41	0.227	0.091	0.650	0.091	0.182	0.410	0.045
Canard	—	0.656	0.288	0.115	0.038	0.558	0.038	0.115	0.192	0.038
Utility System										
Gun	—	—	—	—	—	0.04	—	—	—	—
Steering	—	0.024	0.024	—	—	—	—	—	—	0.24
Radar	—	0.024	0.024	0.024	0.024	0.024	0.024	0.024	0.024	0.024
Nozzles	—	2.37	2.179	0.077	0.023	0.061	0.023	0.015	0.405	0.020
Ramps	—	—	0.002	0.003	0.003	0.080	0.003	0.002	0.002	—
Aileron	—	0.171	0.026	0.026	0.017	0.123	0.017	0.017	0.057	0.008
Flaperon	—	0.656	—	0.103	—	0.037	—	—	0.092	0.11

1 = leakage

Note: Flows are mean values over flight phase in GPM

Figure 219
F-15 SMTD
 Operational Flow 3,000 psi Intelligent Pump

Per Actuator	Null Leakage (per Aircraft)	Flight Phase								
		Taxi	STOL	Clmb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	0.088	0.088	0.061	0.061	0.07	—	0.07	0.07	0.03	0.079
Flaperon	0.35	0.35	0.35	—	0.35	0.016	0.35	0.35	0.035	0.315
Rudder	0.053	0.053	0.037	0.047	0.047	—	0.047	0.047	—	—
Stabilator	0.70	0.70	0.07	0.35	0.56	0.14	0.56	0.42	0.07	0.63
Canard	0.70	0.70	0.175	0.49	0.63	0.56	0.63	0.49	0.35	0.63
Utility System										
Gun	0.61	0.61	0.61	0.61	0.61	—	0.61	0.61	0.61	0.61
Steering	0.08	0.02	0.02	—	—	—	—	—	—	0.02
Radar	—	—	—	—	—	—	—	—	—	—
Nozzles	2.80	2.80	—	—	—	—	—	1.40	2.54	2.80
Ramps	1.05	1.05	1.043	1.040	1.040	0.858	1.040	1.040	1.043	1.05
Aileron	0.175	0.175	0.123	0.123	0.14	—	0.14	0.14	0.058	0.158
Flaperon	0.70	0.70	0.70	—	0.70	0.70	0.70	0.70	0.70	0.70

Note: Flows are mean values over flight phase in GPM

Figure 220
F-15 SMTD
 Leakage Flow 3,000 psi Intelligent Pump

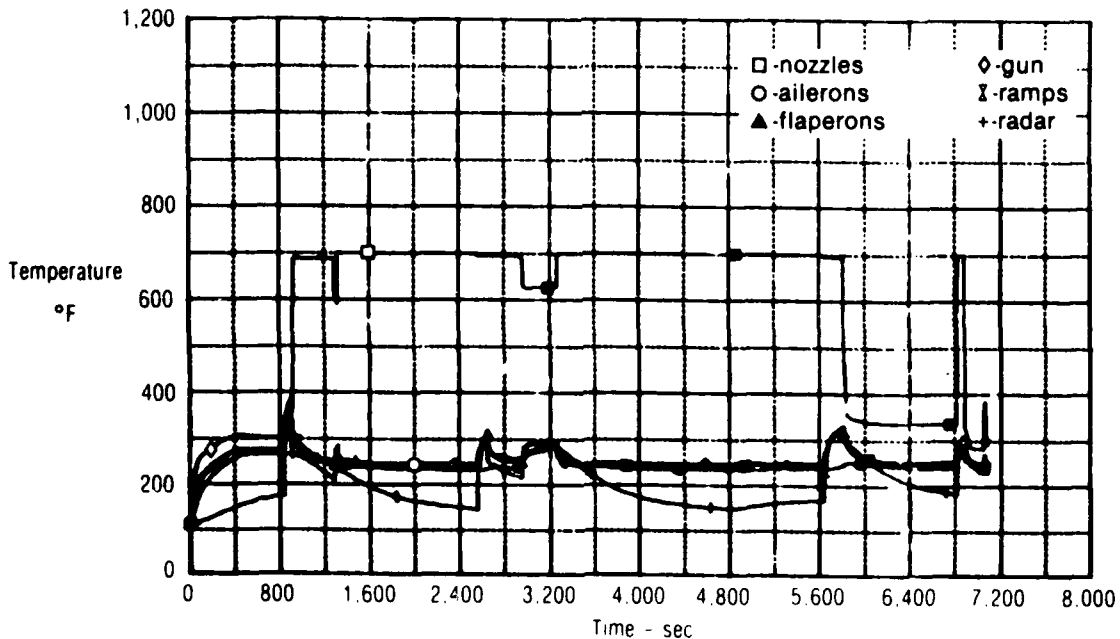


Figure 221
8,000 psi Intelligent Pump Without Ram Air HX
 Utility Actuator Exit Temperatures

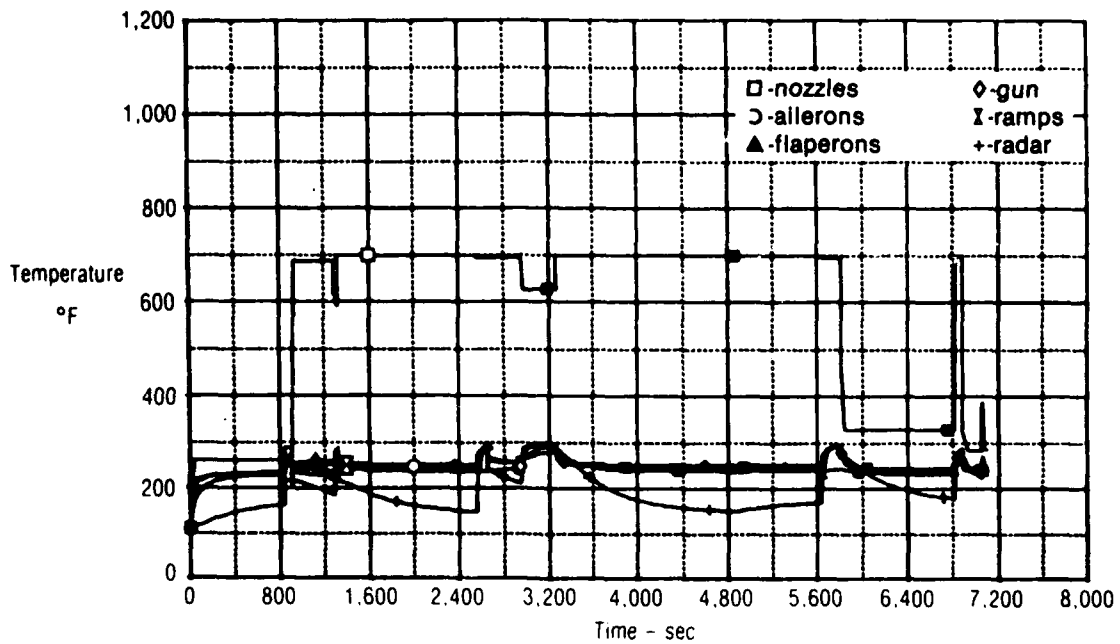


Figure 222
8,000 psi Intelligent Pump
 Utility Actuator Exit Temperatures

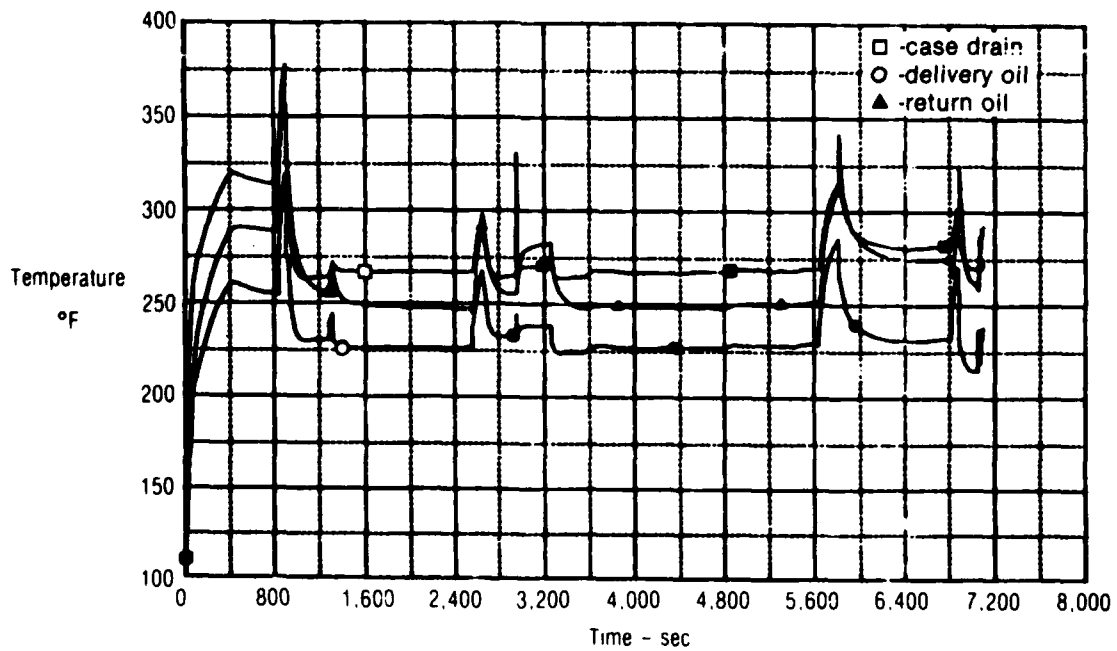


Figure 223
8,000 psi Intelligent Pump Without Ram Air HX
 Utility Pump and Return Temperatures

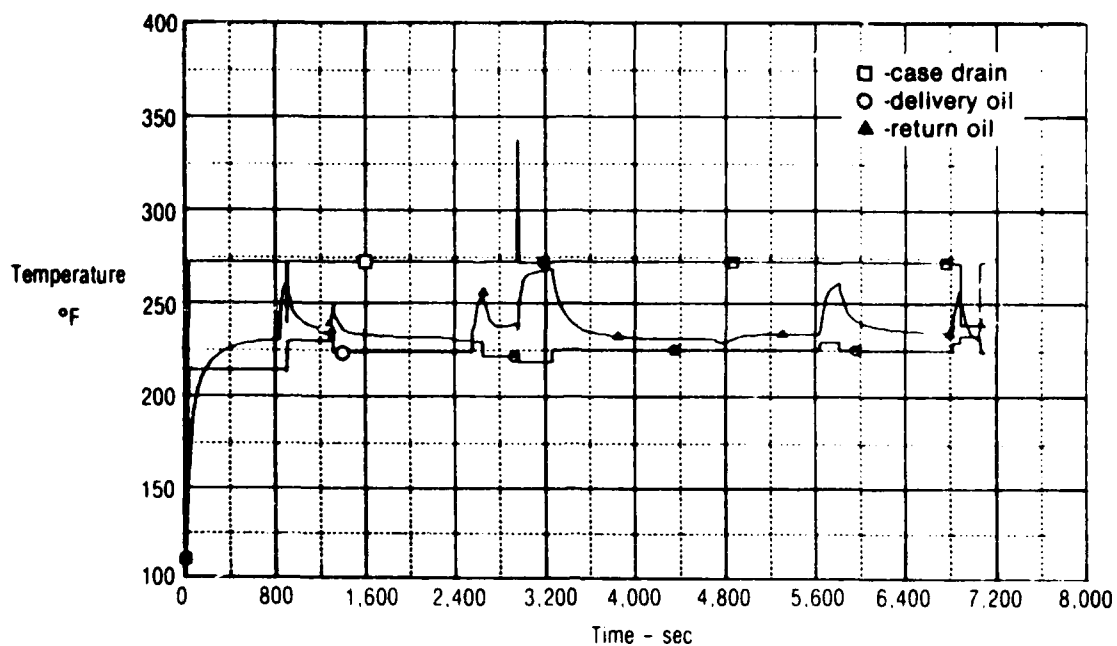


Figure 224
8,000 psi intelligent Pump
 Utility Pump and Return Temperatures

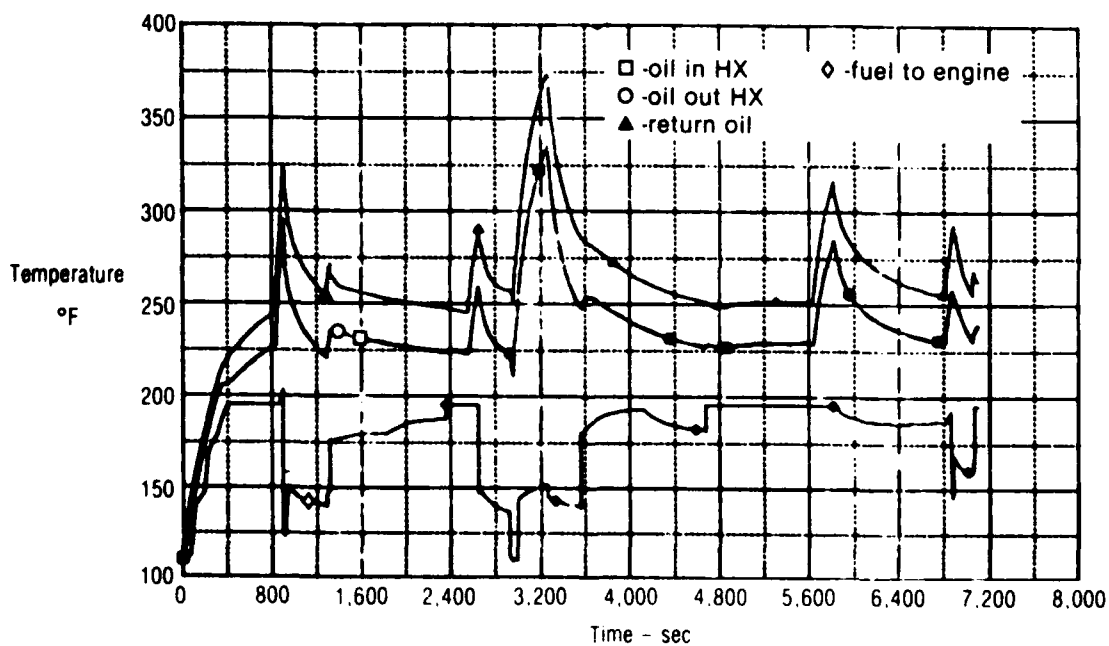


Figure 225
8,000 psi Intelligent Pump Without Ram Air HX
 Utility Ram Air HX Requirements

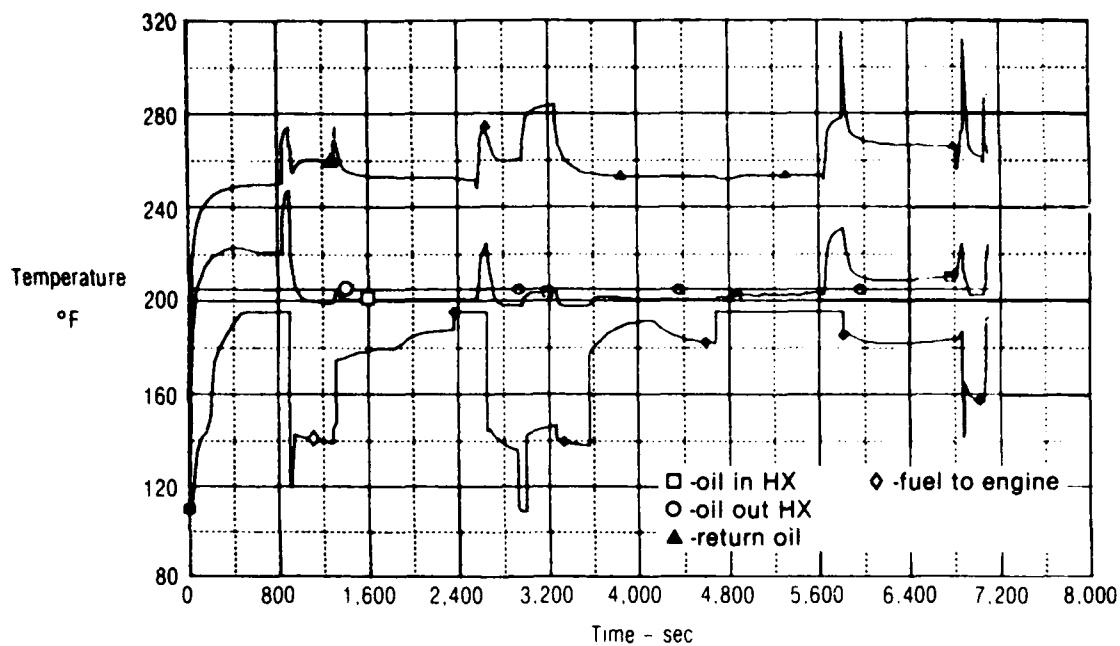


Figure 226
8,000 psi Intelligent Pump
Utility Ram Air HX Requirements

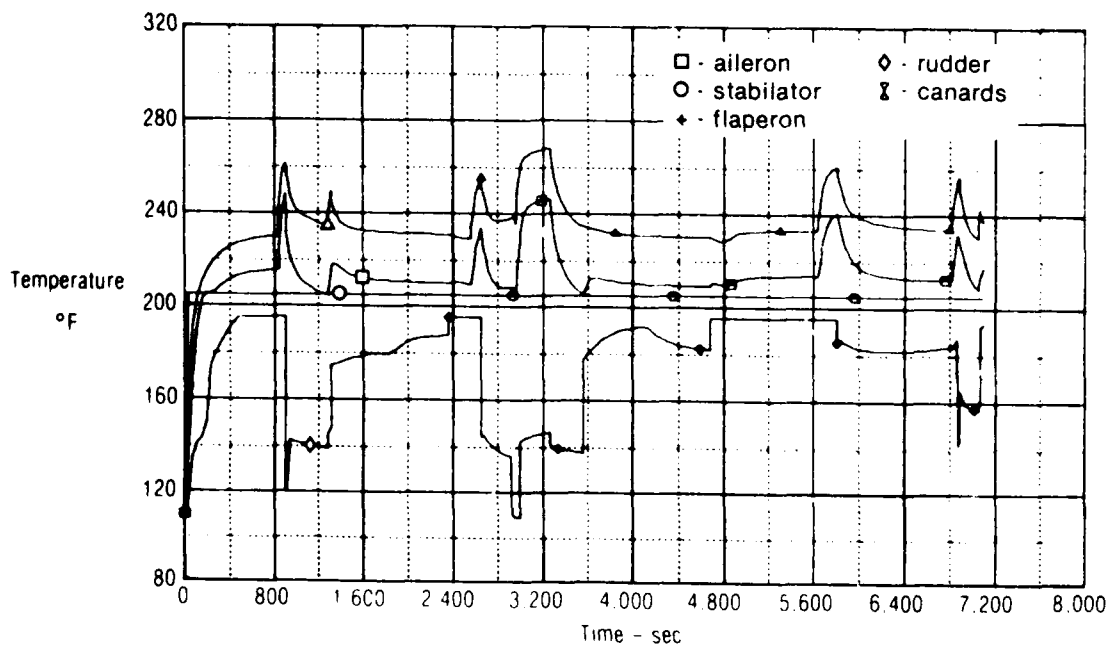


Figure 227
8,000 psi Intelligent Pump Without Ram Air HX
PC-1 Actuator Exit Temperatures

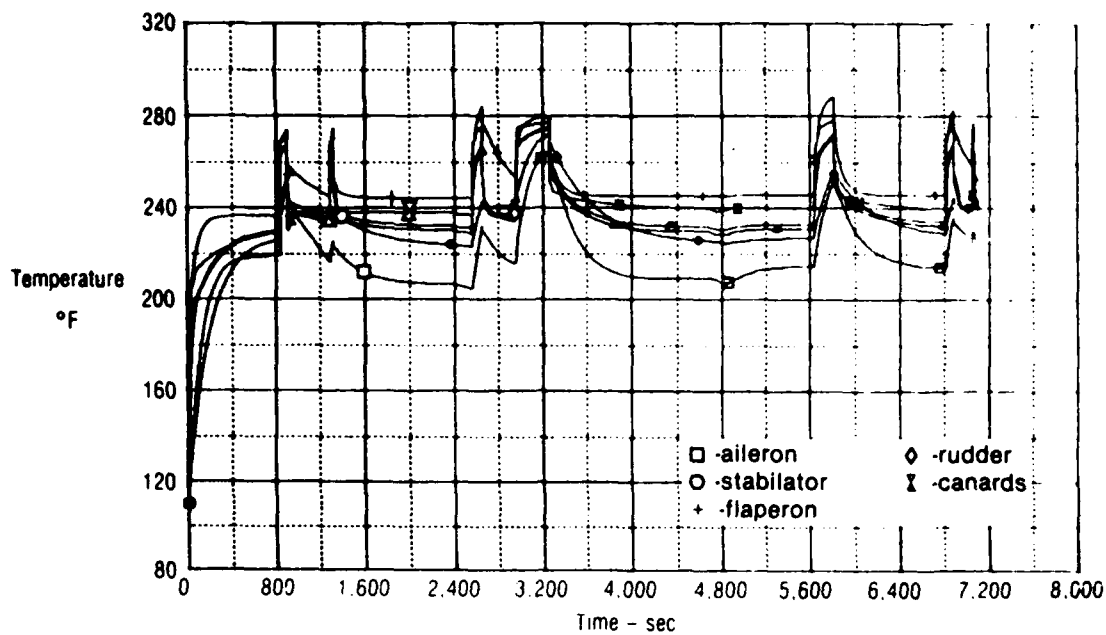


Figure 228
8,000 psi Intelligent Pump
PC-1 Actuator Exit Temperatures

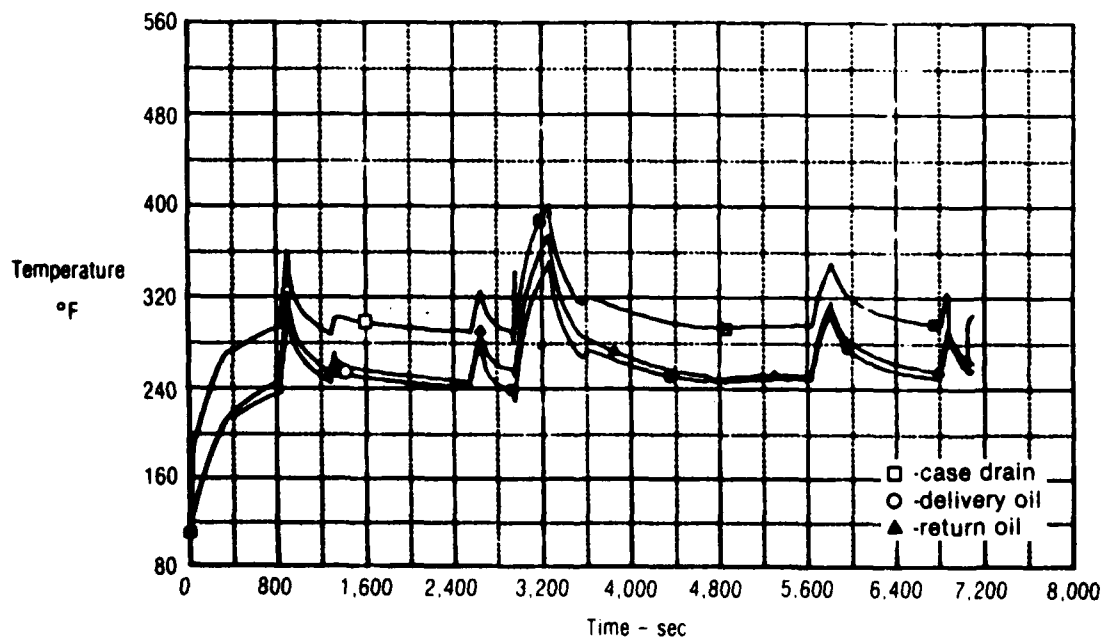


Figure 229
8,000 psi Intelligent Pump Without Ram Air HX
PC-1 Pump and Return Temperatures

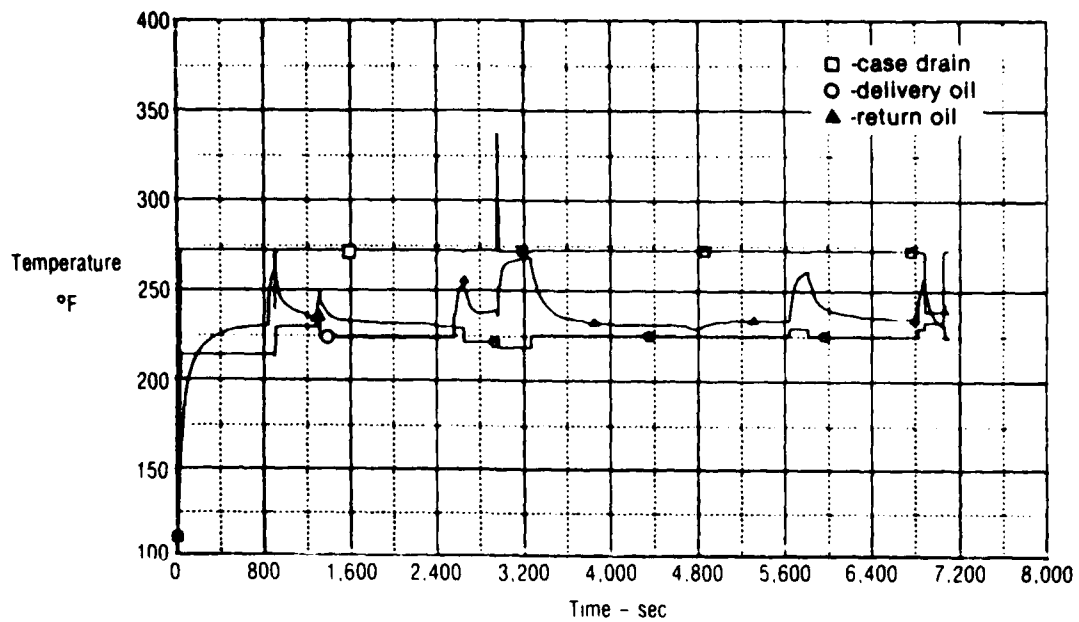


Figure 230
8,000 psi Intelligent Pump
PC-1 Pump and Return Temperatures

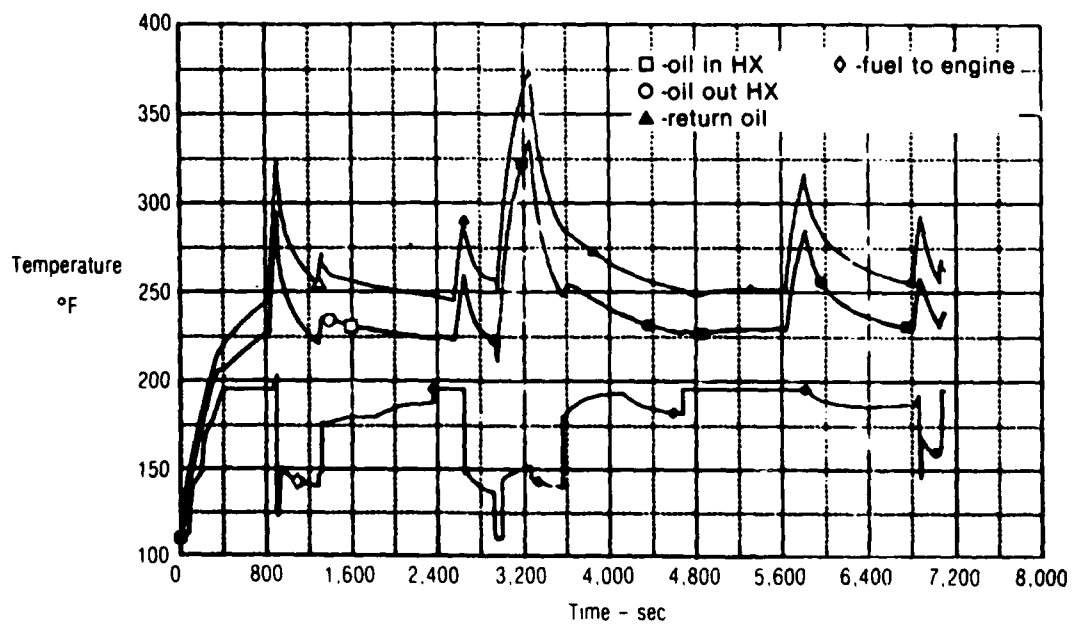


Figure 231
8,000 psi Intelligent Pump Without Ram Air HX
PC-1 Ram Air HX Requirements

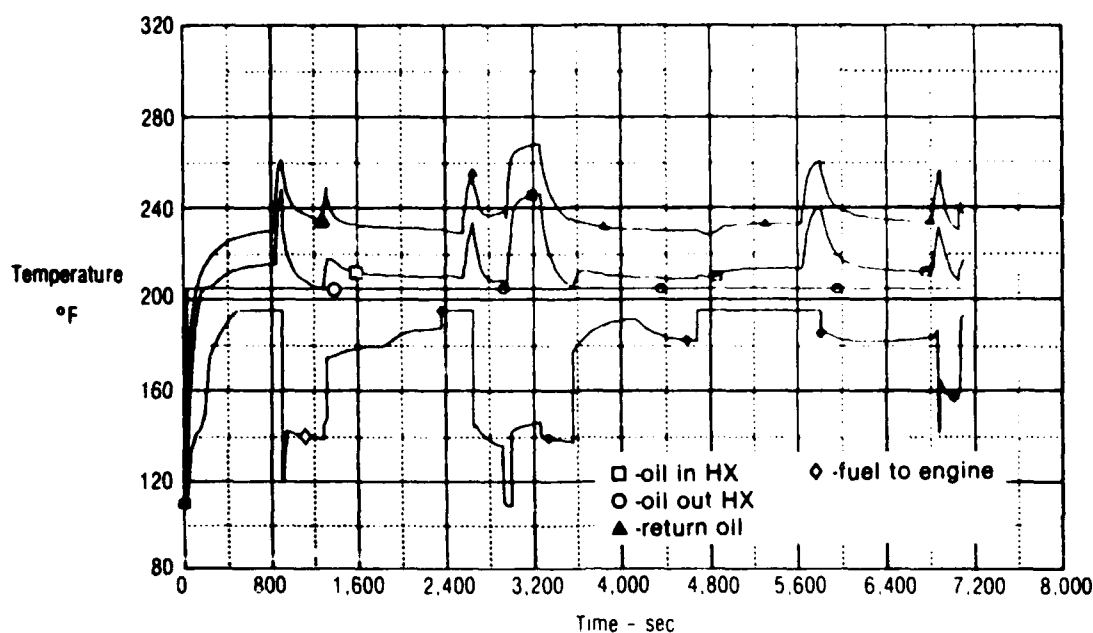


Figure 232
8,000 psi Intelligent Pump
PC-1 Ram Air HX Requirements

Concept	Sys	HX	Fan	Motor	Duct	Install	Total	Total HX	ΔWeight
8,000 psi Baseline	Utility	13.4	16.4	35.4	9.0	18.6	92.8	174.4	—
		13.7	14.2	29.9	7.5	16.3	81.6		
Intelligent Pump	Utility	8.6	3.6	8.6	7.2	7.0	35.0	59.4	- 115.0
		2.4	4.8	9.5	2.8	4.9	24.4		

Fuel to oil heat exchanger weighs 9.1 lb per aircraft engine

Figure 233
Intelligent Pump
Ram Air Heat Exchanger Weight Breakdown - Lb

b. Distribution System and Component Weight - Figure 234 shows the hydraulic system weight summary for the intelligent pump concept. Relative to the baseline weight, differences occurred with the miscellaneous components and the ram air heat exchangers. The miscellaneous components increased 26 lb. because of the addition of a 10 lb. pressure intensifier on the nozzle actuators, which maintained the holding load requirement when the

	8,000 psi		
	Baseline	Intelligent Pump	ΔWeight
Flight Control Actuators	325.73	325.73	--
Engine Nozzle Actuators	284.36	284.36	—
Utility Actuators	115.40	115.40	—
Miscellaneous Components	436.25	462.25	+ 26.00
Ram Air Heat Exchangers	192.60	77.60	- 115.00
Distribution System	197.92	197.92	—
Fluid - CTFE	181.77	181.77	—
Total	1,734.03	1,645.03	- 89.00

Note: Ram Air Heat Exchanger weights were established from thermal analyses includes 18.2 lb fuel/oil exchanger weight.

Figure 234
Intelligent Pump Hydraulic System Weight Summary - Lb

system was at 3,000 psi. The ram air heat exchanger weight decreased 115 lb. because of the reduced system heat rejection. The total hydraulic system weight decreased 89 lb. Figure 235 shows the detailed component weight breakdown for the 8,000 psi intelligent pump concept.

c. Life Cycle Costs - The intelligent pump concept LCCs were analyzed and quantified relative to the 8,000 psi baseline configuration.

The hydraulic system component affected by the intelligent pumps is the system pump. As shown in Figure 236, the pump exhibited a decrease in reliability and an increase in maintainability, based upon data received from pump suppliers.

Figure 237 shows a summary of the hydraulic and aircraft systems LCC.

The affect on total aircraft LCCs of arbitrarily increasing the Mean-Time-Between-Failures (MTBF) value of each hydraulic system component, was evaluated with the intelligent pump system only. As shown, the hydraulic system unit flyaway costs decreased \$4,000. The total hydraulic system LCCs for the intelligent pump system increased \$7 M, the 2XMTBF LCCs decreased \$75 M and the 3XMTBF LCCs decreased \$99 M.

When the hydraulic system was analyzed with the total aircraft system, the 1XMTBF costs decreased \$142 M, the 2XMTBF decreased \$224 M and the 3XMTBF decreased \$248 M.

	Dry Weight (lb)	Fluid Volume (in. ³)	Fluid Weight (lb) CTFE	Wet Weight (lb)
Nozzle Controls				
Actuators				
Upper Rotating Vane	24.20	7.45	0.48	24.68
Lower Rotating Vane	24.20	7.45	0.48	24.68
Outboard Divergent Flap	84.52	56.04	3.59	88.11
Lower Divergent Flap	84.52	56.04	3.59	88.11
Convergent Flap	66.92	60.68	3.88	70.80
Valves	19.66	7.81	0.50	20.16
Total	304.02	195.47	12.52	316.54
Hydraulic PC-1 and PC-2				
Valve				
Temperature Regulator	4.08	12.50	0.80	4.68
Other	0.40			0.40
Miscellaneous				
Pump	72.00	80.00	5.12	69.12
Reservoir	18.60	224.00	14.34	32.94
Other	40.44	25.00	1.60	42.04
Total	135.52	341.50	21.86	157.38
Hydraulic Utility System				
Valve				
Temperature Regulator	2.04	6.25	0.40	2.44
Other	4.50			4.50
Miscellaneous				
Pump	72.00	80.00	5.12	69.12
Reservoir	22.90	348.00	22.27	45.17
Primary Heat Exchanger	0.83	0.63	0.04	0.87
Primary HX Valve	1.17			1.17
Other	41.73	43.44	2.78	44.51
Total	145.17	478.32	30.61	175.78

Ram Air Heat Exchanger Requirements			
	PC-1 + PC-2 Systems (lb)	Utility System (lb)	Total (lb)
Fuel/Oil HX	9.1	9.1	18.2
Ram Air HX	8.6	2.4	11.0
Fans	3.6	4.8	8.4
Electric Motor	8.6	9.5	18.1
Duct	7.2	2.8	10.0
Installation	7.0	4.9	11.9
Total	44.1	33.5	77.6

Figure 235
Intelligent Pump
Hydraulic Equipment Weight Changes - Lb

Concept	Components Affected	Type	Quantity Aircraft	Reliability MFHBF	Maintainability MTTR
Intelligent Pump	Pump	—	4	7.820	6.06

Figure 236
Intelligent Pump
Reliability and Maintainability Changes

	8,000 psi	
	Baseline	Intelligent Pump
Hydraulic System Unit Flyaway	—	- 0.004
Hydraulic System Life Cycle Cost	—	+ 7.00 - 75.00 ⁽²⁾ - 99.0
Aircraft System Unit Flyaway	—	- 0.109
Total Aircraft System Life Cycle Cost	—	- 142.0 - 224.0 ⁽²⁾ - 248.0
Total Aircraft Weight (lb)	28,047	27,849
ΔAircraft Weight	—	- 198

Notes:

(1) Life cycle costs are millions of FY dollars

(2) Intelligent Pump Life Cycle Cost depicts a: 1-(MTBF), 2-(MTBF), 3-(MTBF)

Figure 237
Intelligent Pump
Hydraulic and Aircraft System Life Cycle Cost/Weight Summary

This study showed on a preliminary basis that the 2XMTBF yielded the largest percentage decrease in LCCs. Increasing to 3XMTBF yielded little LCC increase over that shown with 2XMTBF.

The total aircraft weight showed a decrease of 198 lb.

3.3.6 Combination of Concepts - The final configuration analyzed combined all concepts into one system. The concepts incorporated were:

- o Overlap valves
- o Flow augmentation/load recovery valves
- o Dry sump pump
- o Intelligent pump

The system pumps were multipressure (3,000 to 8,000 psi), and dry sumped, with a lower output flow rate. All valves incorporated 0.010 inch overlap on the spools. The flight control actuators incorporated the flow augmentation/load recovery valve concept. Detailed descriptions of each low energy consumption concept are located in Sections 3.3.2, 3.3.3, 3.3.4 and 3.3.5.

a. Thermal Analysis - The thermal analysis for the combination of concepts was divided into two segments. The first segment was the determination of the pump heat rejection and the other was the determination of system average operational and average leakage flow rates.

The system pump heat rejection is shown in Figure 101 for the combination system. There are two levels of heat rejection. The 8,000 psi pump heat rejection is 31 horsepower per aircraft or 7.75 horsepower per pump. Decreasing the system pressure to 3,000 psi, decreased the pump heat rejection to 15.4 horsepower per aircraft or 3.87 horsepower per pump.

Figure 238 shows the thermal parameters established for the pump. The case drain flow was 1.0 gpm at 8,000 psi and 0.7 gpm at 3,000 psi. The heat rejection into the case drain and pump discharge, remained in the two-thirds/one-third relationship for each pressure.

Configuration	Pump Heat Rejection (hp)	H.R. Accounted for (hp)	Case Drain Flow (gpm)	H.R. to Case Drain (hp)	H.R. to Discharge (hp)	Actuator Operational Flows	Actuator Leakage Flows	System Pressure (psi)
8,000 psi Baseline	16.64	14.14 (85%)	2.0	9.47 (67%)	4.67 (33%)	Figure 122	Figure 123	8,000
Combinations	9.11/4.56 (8,000/3,000)	7.74/3.87 (85%)	1.0/0.7	5.16/2.59 (67%)	2.55/1.28 (33%)	Figure 239	Figure 240	8,000/3,000

Figure 238
Combination Concept Thermal Model Parameters

The actuator operational and leakage flow rates are shown in Figures 239 through 242. The flow rates are tabulated for 3,000 and 8,000 psi system operation. The operational flows are the same used in Section 3.3.5. The leakage flows were based on 0.010 inch valve overlap, and are an order of magnitude lower than the leakage flows used in Section 3.3.5. The flows were segmented as described in Section 3.3.5.a.

Per Actuator	Actuator Flow (Unloaded)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	1.08	0.228 ¹	0.034	0.034	0.023	0.164	0.023	0.023	0.076	0.011
Flaperon	1.31	0.875 ¹	—	0.137	—	0.051	—	—	0.122	0.014
Rudder	1.37	0.175 ¹	0.043	0.015	0.015	0.208	0.015	0.015	0.143	0.143
Stabilator	11.60	1.75 ¹	1.092	0.605	0.242	1.732	0.242	0.484	1.092	0.121
Canard	9.81	1.75 ¹	0.768	0.307	0.102	1.489	0.102	0.307	0.512	0.102
Utility System										
Gun	10.88	—	—	—	—	0.107	—	—	—	—
Steering	0.90	0.64 ¹	0.64	—	—	—	—	—	—	0.64
Radar	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64	0.64
Nozzles	—	7.28	5.81	0.204	0.061	0.162	0.061	0.039	1.08	0.54
Ramps	5.91	—	0.006	0.008	0.008	0.213	0.008	0.006	0.006	—
Aileron	2.17	0.456 ¹	0.069	0.069	0.046	0.329	0.046	0.046	0.152	0.022
Flaperon	2.61	1.75 ¹	—	0.274	—	0.099	—	—	0.244	0.028

¹ = leakage

Note: Flows are mean values over flight phase in GPM

Figure 239
F-15 SMTD
 Operational Flow 8,000 psi Combination of Concepts

Per Actuator	Null Leakage (Per Aircraft)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	0.013	0.013	0.0092	0.0092	0.0104	—	0.0104	0.0104	0.0042	0.0118
Flaperon	0.05	0.05	0.05	—	0.05	0.05	0.05	0.05	—	0.045
Rudder	0.01	0.01	0.01	0.007	0.009	—	0.009	0.009	—	—
Stabilator	0.10	0.10	0.01	0.05	0.08	0.02	0.08	0.06	0.01	0.09
Canard	0.10	0.10	0.10	0.025	0.07	0.09	0.09	0.07	0.05	0.09
Utility System										
Gun	0.034	0.034	0.034	0.034	0.034	—	0.034	0.034	0.034	0.034
Steering	0.011	0.0032	0.0032	—	—	—	—	—	—	0.0032
Radar	—	—	—	—	—	—	—	—	—	—
Nozzles	0.416	0.416	—	—	—	—	—	0.208	0.3786	0.416
Ramps	0.156	0.156	0.155	0.1528	0.1528	0.1326	0.1528	0.155	0.155	0.144
Aileron	0.026	0.026	0.0182	0.0208	—	—	—	0.0208	0.0086	0.0234
Flaperon	0.10	0.10	0.10	—	0.10	0.10	0.10	0.10	—	0.09

Note: Flows are mean values over flight phase in GPM

Figure 240
F-15 SMTD
Leakage Flow 8,000 psi Combination of Concepts

Per Actuator	Actuator Flow (Unloaded)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	—	0.086	0.013	0.013	0.009	0.062	0.009	0.009	0.029	0.004
Flaperon	—	0.328	—	0.051	—	0.019	—	—	0.046	0.005
Rudder	—	0.066	0.016	0.006	0.006	0.078	0.006	0.006	0.054	0.054
Stabilator	—	0.656	0.41	0.227	0.091	0.650	0.091	0.182	0.410	0.045
Canard	—	0.656	0.288	0.115	0.038	0.558	0.038	0.115	0.192	0.038
Utility System										
Gun	—	—	—	—	—	0.04	—	—	—	—
Steering	—	0.024	0.024	—	—	—	—	—	—	0.24
Radar	—	0.024	0.024	0.024	0.024	0.024	0.024	0.024	0.024	0.024
Nozzles	—	2.37	2.179	0.077	0.023	0.061	0.023	0.015	0.405	0.020
Ramps	—	—	0.002	0.003	0.003	0.080	0.003	0.002	0.002	—
Aileron	—	0.171	0.026	0.026	0.017	0.123	0.017	0.017	0.057	0.008
Flaperon	—	0.656	—	0.103	—	0.037	—	—	0.092	0.11

1 = leakage

Note: Flows are mean values over flight phase in GPM

Figure 241
F-15 SMTD
Operational Flow 3,000 psi Combination of Concepts

Per Actuator	Null Leakage (Per Aircraft)	Flight Phase								
		Taxi	STOL	Climb and Retract Gear	Cruise	Combat	Cruise	Descent	STOL	Reverse
PC-1 and PC-2 Systems										
Aileron	0.0088	0.0088	0.0061	0.0061	0.0061	0.007	0.007	0.007	0.003	0.0079
Flaperon	0.035	0.035	0.035	—	0.035	0.0016	0.035	0.035	0.035	0.0315
Rudder	0.0053	0.0053	0.0037	0.0047	0.0047	—	0.0047	0.0047	—	—
Stabilator	0.070	0.070	0.007	0.035	0.056	0.014	0.056	0.042	0.007	0.063
Canard	0.070	0.070	0.0175	0.049	0.063	0.056	0.063	0.049	0.035	0.063
Utility System										
Gun	0.061	0.061	0.061	0.061	0.061	—	0.061	0.061	0.061	0.061
Steering	0.008	0.002	0.002	—	—	—	—	—	—	0.002
Radar	—	—	—	—	—	—	—	—	—	—
Nozzles	0.0281	0.280	—	—	—	—	—	0.140	0.254	0.280
Ramps	0.105	0.105	0.1043	0.1040	0.1040	0.0858	0.1040	0.1040	0.1043	0.105
Aileron	0.0175	0.0175	0.0123	0.0123	0.014	—	0.014	0.014	0.0058	0.0158
Flaperon	0.070	0.070	0.070	—	0.070	0.070	0.070	0.070	0.070	0.070

Note: Flows are mean values over flight phase in GPM

Figure 242
F-15 SMTD
Leakage Flow 3,000 psi Combination of Concepts

Figures 243 through 254 show the temperatures computed when the above flows and pump heat rejection were used with the thermal computer model, as mentioned in Section 3.1.2.a. These plots should be interpreted as described in Section 3.3.1.a.

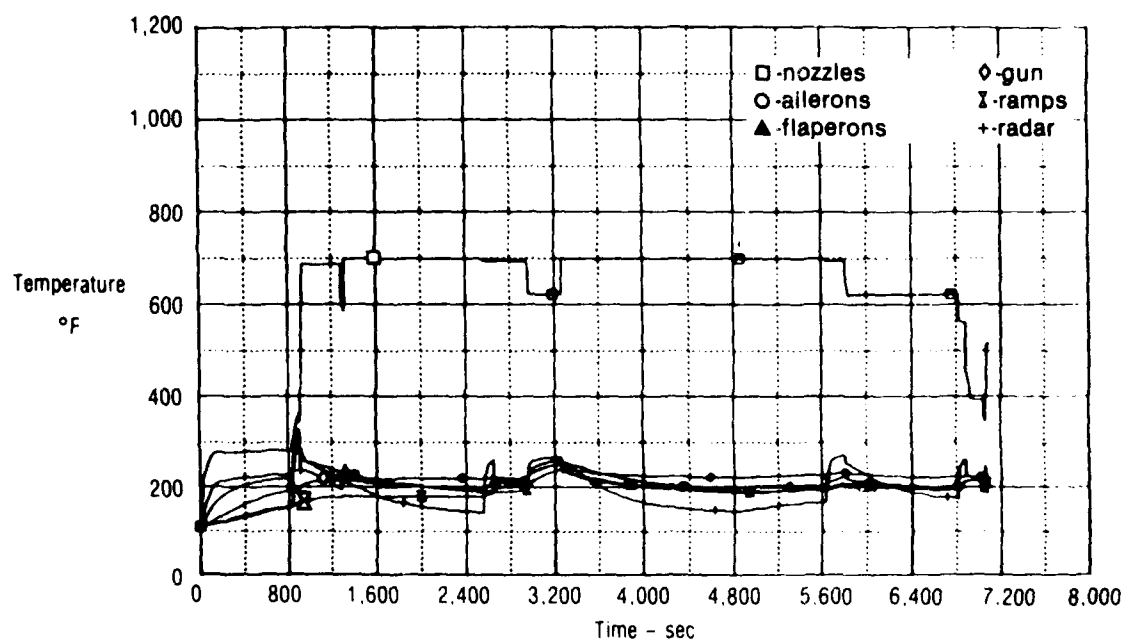


Figure 243
8,000 psi Combination Concepts Without Ram Air HX
Utility Actuator Exit Temperatures

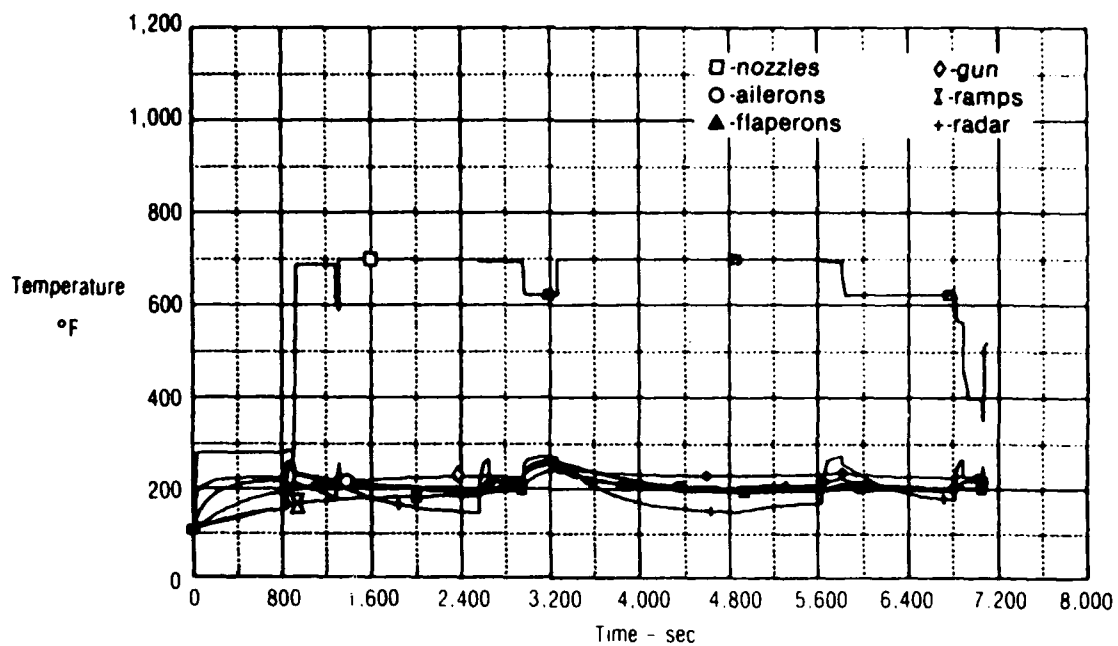


Figure 244
8,000 psi Combination Concepts
Utility Actuator Exit Temperatures

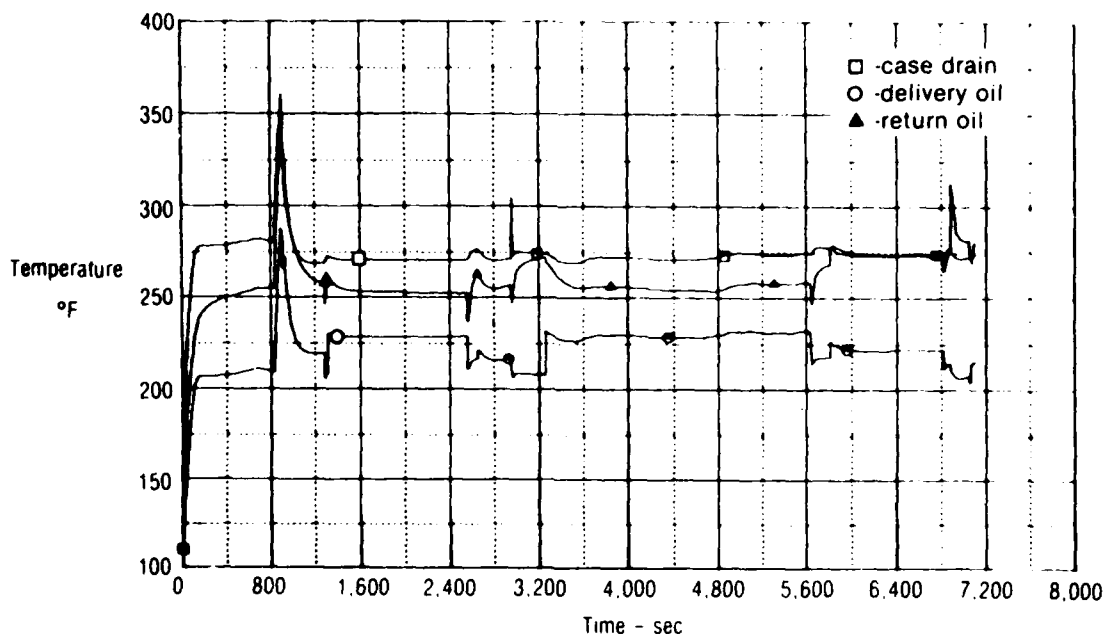


Figure 245
8,000 psi Combination Concepts Without Ram Air HX
Utility Pump and Return Temperatures

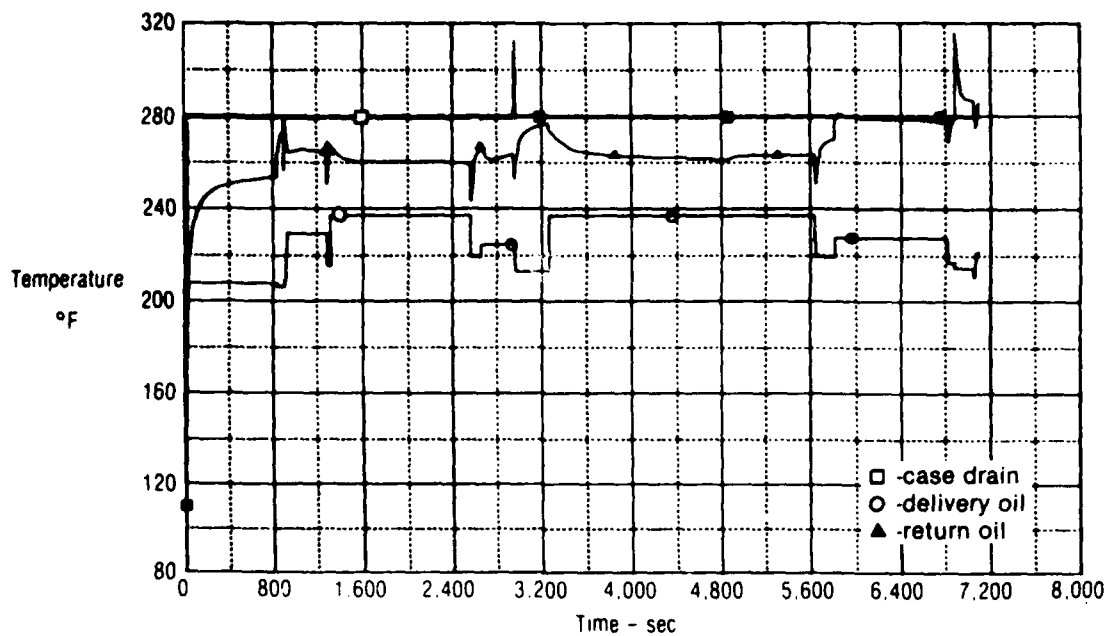


Figure 246
8,000 psi Combination Concepts
Utility Pump and Return Temperatures

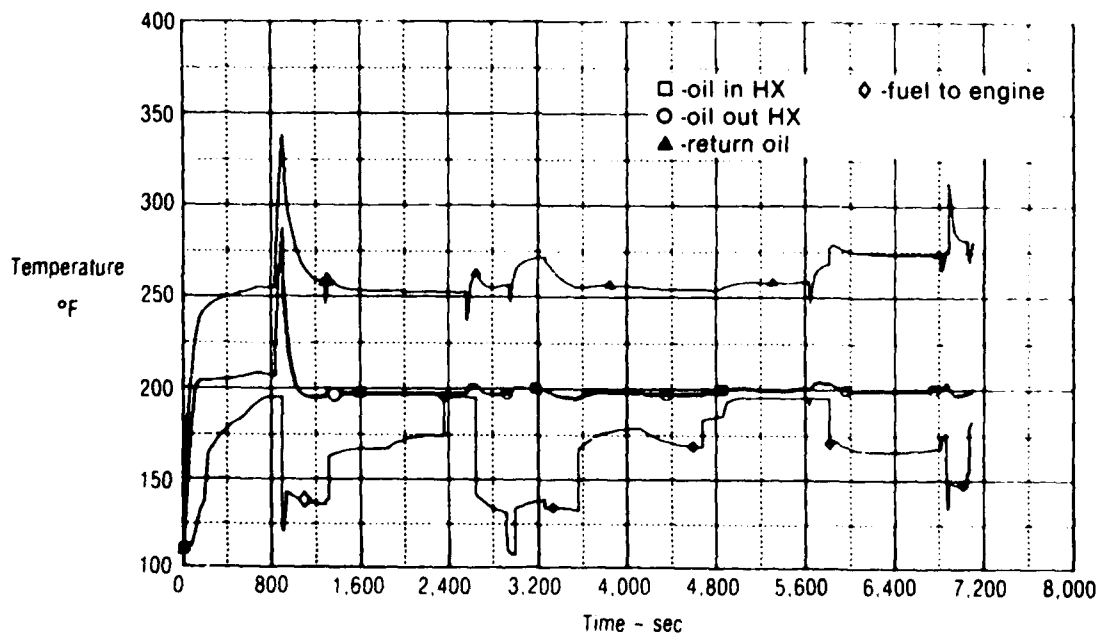


Figure 247
8,000 psi Combination Concepts Without Ram Air HX
Utility Ram Air HX Requirements

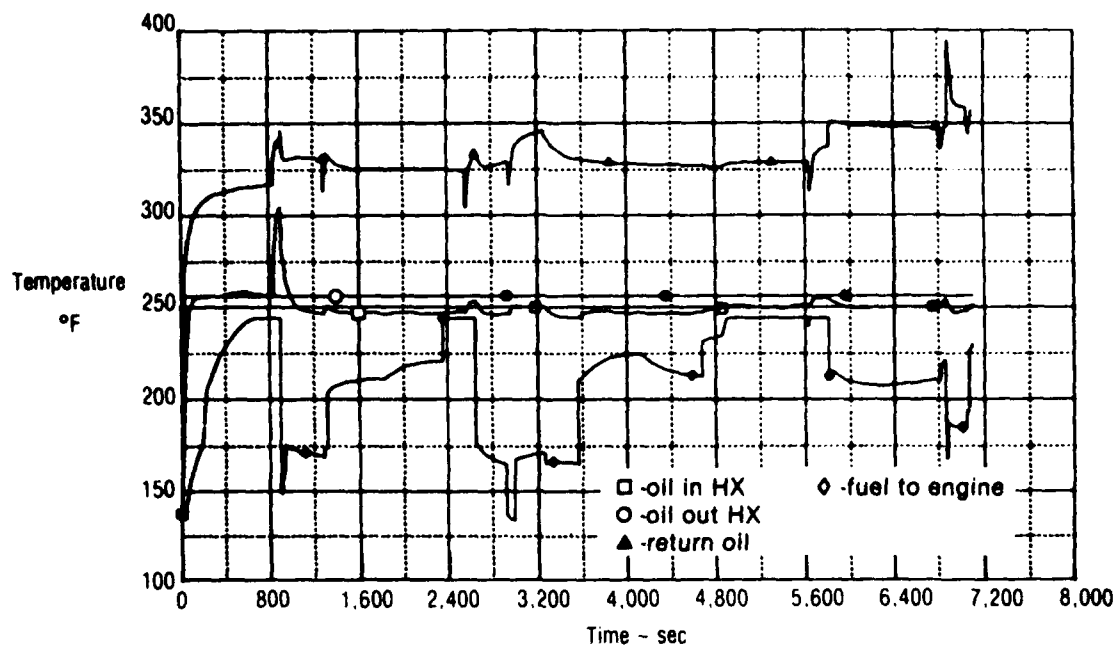


Figure 248
8,000 psi Combination Concepts Pump
Utility Ram Air HX Requirements

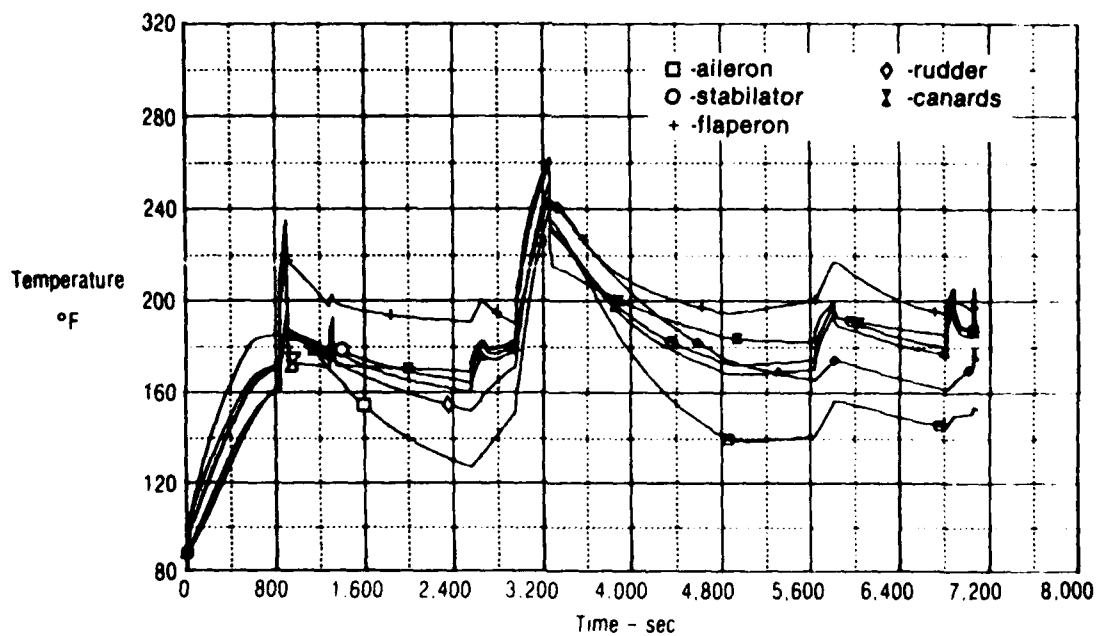


Figure 249
8,000 psi Combination Concepts Without Ram Air HX
PC-1 Actuator Exit Temperatures

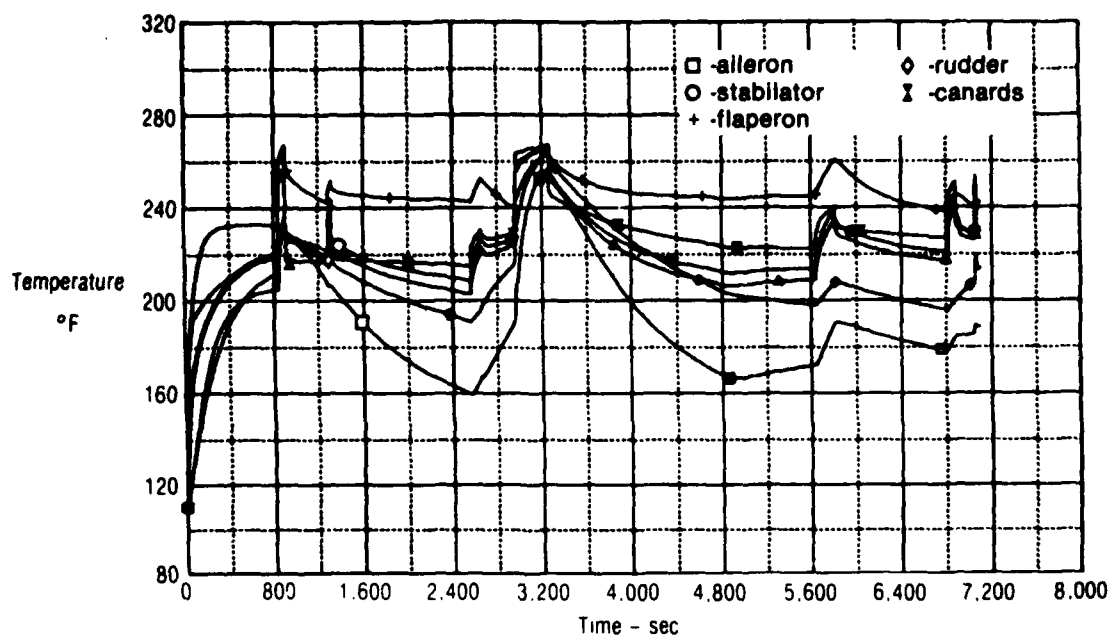


Figure 250
8,000 psi Combination Concepts
PC-1 Actuator Exit Temperatures

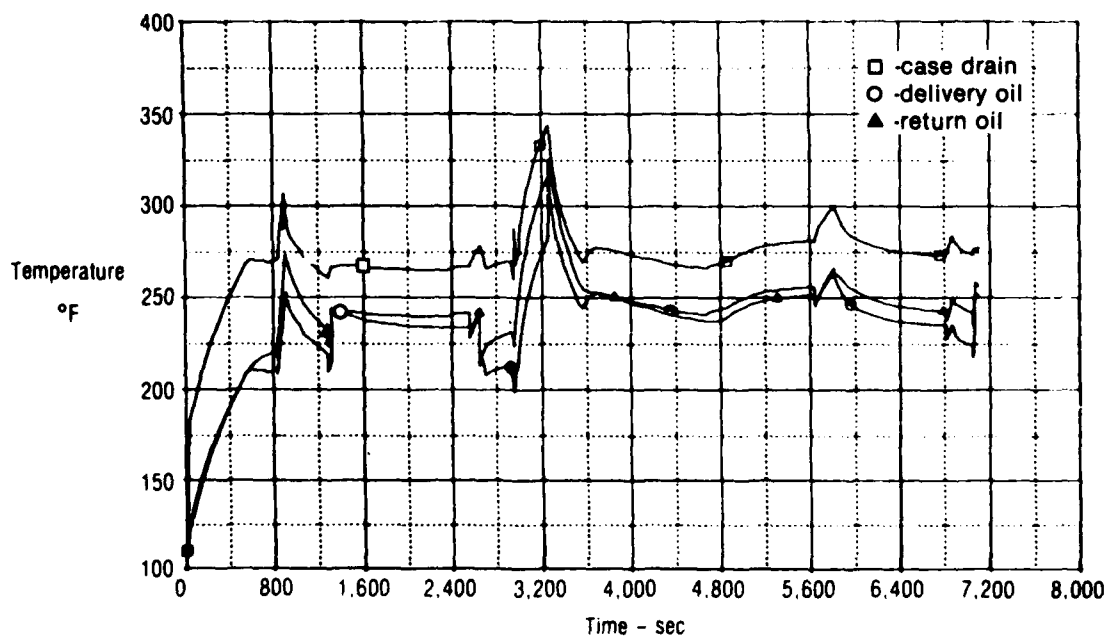


Figure 251
8,000 psi Combination Concepts Without Ram Air HX
PC-1 Pump and Return Temperatures

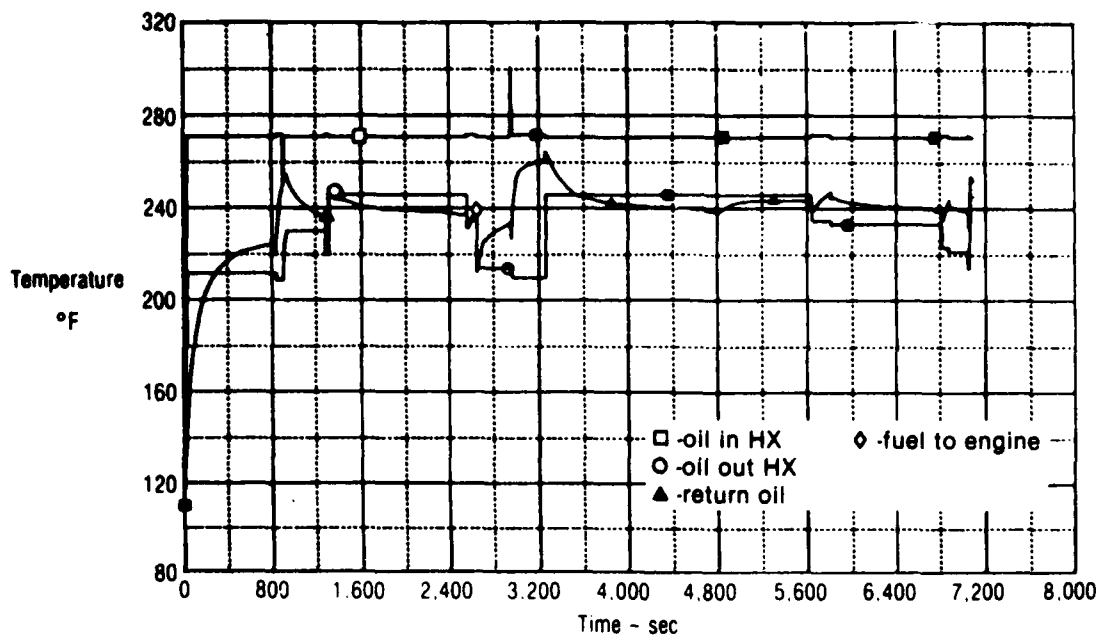


Figure 252
8,000 psi Combination Concepts
PC-1 Pump and Return Temperatures

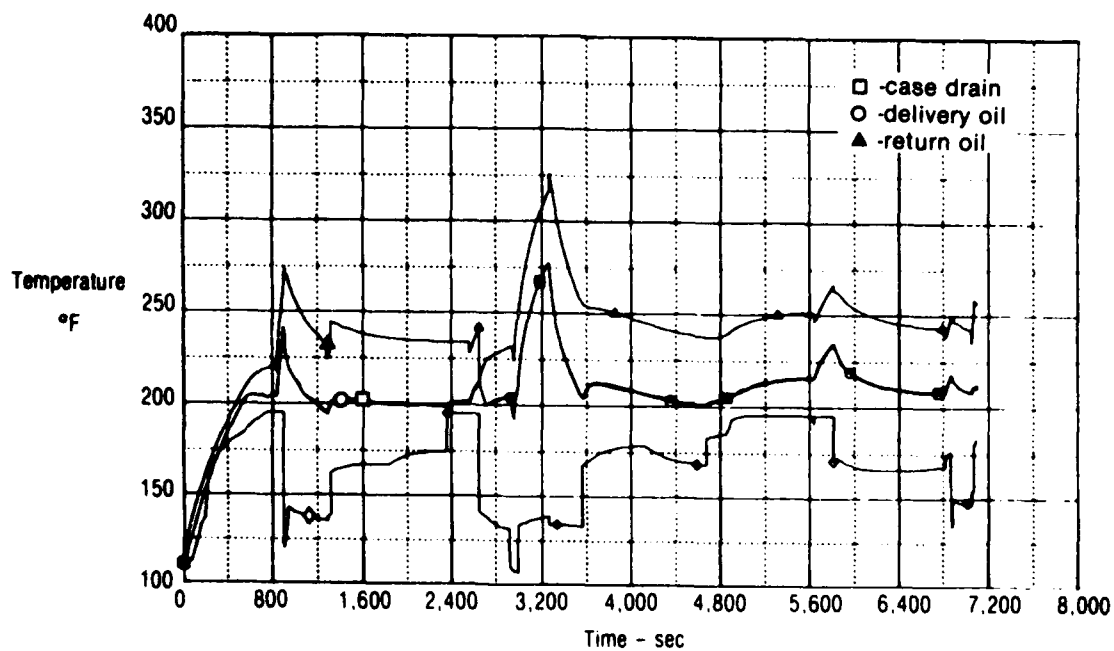


Figure 253
8,000 psi Combination Concepts Without Ram Air HX
PC-1 Ram Air HX Requirements

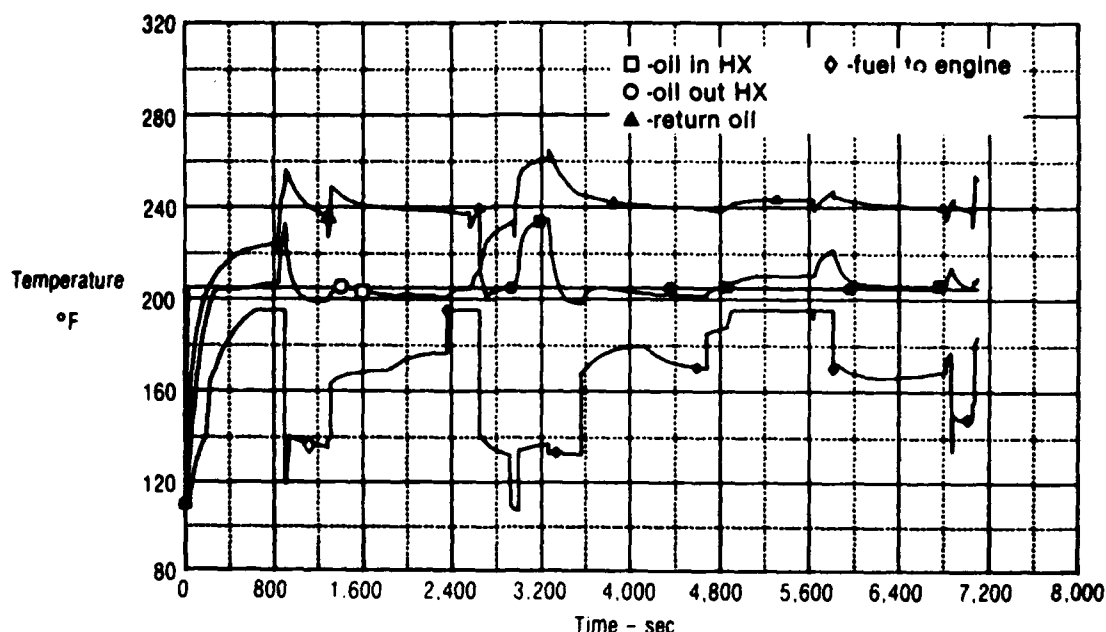


Figure 254
8,000 psi Combination Concepts
PC-1 Ram Air HX Requirements

The weight impact of the ram air heat exchanger when utilizing the combination concept is shown in Figure 255. The analysis and sizing techniques for the ram air heat exchangers are described in Section 3.3.1.a.

The PC heat exchangers for the combination of concepts, were sized by the combat flight phase because of the increased heat load at 8,000 psi. As indicated in Figure 247, the utility system remained below 275°F except for approximately 150.0 seconds during taxi. It was decided that this temperature transient above 275°F would not be detrimental to the system, so no additional ram air heat exchangers were required. Figure 253 shows that the PC system will require additional cooling capacity for the combat flight phase.

Figure 255 shows that the heat exchanger weights decreased 166.8 lb. from the baseline. Since the maximum temperature did not occur during taxi, no fans or motors were required.

b. Distribution System and Component Weight - Figure 256 shows the hydraulic system weight summary for the combination of concepts. Relative to the baseline weight, differences occurred with the flight control actuators, ram air heat exchangers, distribution system, miscellaneous components and the CTFE fluid.

The flight control actuators showed an increase from the baseline of 19.08 lb. due to flow augmentation/load recovery valves. The rationale is shown in Section 3.3.3.

Concept	Sys	HX	Fan	Motor	Duct	Install	Total	Total HX	ΔWeight
8,000 psi Baseline	PC-1 + PC-2	13.4	16.4	35.4	9.0	18.6	92.8	174.4	—
	Utility	13.7	14.2	29.9	7.5	16.3	81.6		
Combination of Concepts	PC-1 + PC-2	2.4	—	—	3.6	1.6	7.6	7.6	- 166.8
	Utility	—	—	—	—	—	0		

Fuel to oil heat exchanger weighs 9.1 lb per aircraft engine

Figure 255
Combination Concepts
Ram Air Heat Exchanger Weight Breakdown - Lb

	8,000 psi		
	Baseline	Combination	ΔWeight
Flight Control Actuators	325.73	344.81	+ 19.08
Engine Nozzle Actuators	284.36	284.36	—
Utility Actuators	115.40	115.40	—
Miscellaneous Components	436.25	429.46	- 6.79
Ram Air Heat Exchangers	192.80	25.80	- 166.80
Distribution System	197.82	190.34	- 7.58
Fluid-CTFE	181.77	161.63	- 20.14
Total	1,734.03	1,551.80	- 182.23

Note: Ram Air Heat Exchanger weights were established from thermal analysis includes 18.2 lb fuel/oil exchanger weight.

Figure 256
Combination Concepts Hydraulic System Weight Summary - Lb

The miscellaneous components decreased 6.79 lb. due to the decreased pump and reservoir size. The ram air heat exchangers decreased 166.8 lb. because of the significantly lowered system heat rejection. The distribution system weight decreased 7.58 lb. due to the flow augmentation concept. Rationale is given in Section 3.3.3.

The fluid weight decreased 20.14 lb. This occurred because the pump was smaller, dry sumping was incorporated and the distribution system had less fluid volume due to flow augmentation.

The total hydraulic system weight decreased 182.23 lb. Figure 257 shows the detailed component weight breakdown for the 8,000 psi combination of concepts.

c. Life Cycle Costs - The LCCs of the combination of concepts were analyzed and quantified relative to the 8,000 psi baseline configuration.

	Dry Weight (lb)	Fluid Volume (in. ³)	Fluid Weight (lb) CTFE	Wet Weight (lb)
Nozzle Concepts				
Actuators				
Upper Rotating Vane	24.20	7.45	0.48	24.68
Lower Rotating Vane	24.20	7.45	0.48	24.68
Outboard Divergent Flap	84.52	56.04	3.59	88.11
Lower Divergent Flap	84.52	56.04	3.59	88.11
Convergent Flap	66.92	60.68	3.88	70.80
Valves	19.66	7.81	0.50	20.16
Total	304.02	195.47	12.52	316.54
Canards				
Actuators	94.74	89.00	5.70	100.44
Valves	9.66	7.81	0.50	10.16
Total	104.40	96.81	6.20	110.60
Ailerons				
Actuators	57.24	13.00	0.83	58.07
Valves				
Switching	9.66	7.81	0.50	10.16
Other	0.58			0.58
Total	67.48	20.81	1.33	68.81
Stabilator				
Actuators	94.74	89.00	5.70	100.44
Valves				
Switching	9.66	7.81	0.50	10.16
Other	2.08			2.08
Total	106.48	96.81	6.20	112.68
Rudder				
Actuator	40.85	17.63	1.13	41.98
Other	3.00			3.00
Total	43.85	17.63	1.13	44.98
Flaperon				
Actuator	57.24	13.00	0.83	58.07
Total	57.24	13.00	0.83	58.07
Hydraulic PC-1 and PC-2				
Valve				
Temperature Regulator	4.08	12.50	0.80	4.68
Other	0.40			0.40
Miscellaneous				
Pump	58.00	15.63	1.00	59.00
Reservoir	15.93	159.63	10.22	26.15
Other	40.44	25.00	1.60	42.04
Total	118.85	212.76	13.62	132.47
Hydraulic Utility System				
Valve				
Temperature Regulator	2.04	6.25	0.40	2.44
Other	4.50			4.50
Miscellaneous				
Pump	58.00	15.63	1.00	59.00
Reservoir	20.78	283.63	18.15	38.93
Primary Heat Exchanger	0.83	0.63	0.04	0.87
Primary HX Valve	1.17			1.17
Other	41.73	43.44	2.78	44.51
Total	129.05	349.58	22.37	151.42
Distribution System				
PC-1 and PC-2	85.08	429.44	27.48	112.56
Utility	105.26	654.89	41.91	147.17
Total	190.34	1,084.33	69.39	249.73

Ram Air Heat Exchanger Requirements			
	PC-1 + PC-2 Systems (lb)	Utility System (lb)	Total (lb)
Fuel/Oil HX	9.1	9.1	18.2
Ram Air HX	2.4	—	2.4
Fans	—	—	—
Electric Motor	—	—	—
Duct	3.6	—	3.6
Installation	1.6	—	1.6
Total	16.7	9.1	25.8

Figure 257
Combination Concepts
 Hydraulic Equipment Weight Changes - Lb

The hydraulic system components changed by the combination of concepts were the flight control actuators and system pumps. As shown in Figure 258, the actuators exhibited a reduced reliability and an increased maintainability. The pumps exhibited an increased reliability and decreased maintainability. This was based upon data received from actuator and pump suppliers.

Concept	Components Affected	Type	Quantity Aircraft	Reliability MFHBF	Maintainability MTTR
Combination	Canard Actuator	SWFM	2	600	6.54
	Stabilator Actuator	SWFM	2	600	6.54
	Aileron Actuator	SWFM	2	860	4.55
	Flaperon Actuator	SWFM	2	860	4.55
	Rudder Actuator	SWFM	2	1,657	6.88
	Pump	—	4	9,705	4.88

Note: SWFM - Simplex With Force Motor

Figure 258
Combination Concepts
Reliability and Maintainability Changes

Figure 259 summarizes the hydraulic and aircraft system LCCs. The hydraulic system unit flyaway cost decreased \$63,000 and the LCC decreased by \$29 M. When the hydraulic system was analyzed with the total aircraft system, the total LCC decreased \$337 M from the baseline.

The total aircraft weight decreased 406.0 lb.

	8,000 psi	
	Baseline	Combination
Hydraulic System Unit Flyaway	—	- 0.063
Hydraulic System Life Cycle Cost	—	- 29.0
Aircraft System Unit Flyaway	—	- 0.276
Total Aircraft System Life Cycle Cost	—	- 337.0
Total Aircraft Weight (lb)	28,047	27,641
Δ Aircraft Weight (lb)	—	- 406.0

Notes:

- (1) Life cycle costs are millions of FY 85 dollars
- (2) Combination life cycle cost depicts 1-(MTBF)

Figure 259
Combination Concepts
Hydraulic and Aircraft System Life Cycle Cost/Weight Summary

3.4 TASK 2 - LABORATORY TEST SYSTEM

The laboratory test system was designed to verify component/concept performance, measure the input energy to the system and perform limited durability testing of the concepts. The proposed test setup shown in Figure 3, utilized the test circuit designed for the "Flight Worthiness of Fire Resistant Systems" 750 hour demonstration test. The setup was modified as required to satisfy LECHT testing objectives. Fluid temperatures were varied over the temperature range of -65°F to 275°F. Components were tested individually first, then together.

3.4.1 Test Setup and Evaluation - Components were evaluated first without the low energy consumption concepts, and then with them. The stabilator actuator was cycled for a time span with frequencies and magnitudes representative of the production actuator. Pump input torque was measured vs. time at specified fluid and ambient temperatures to determine the energy consumed. In addition, other tests such as comparison of overlap valve and line-to-line valve null leakages, were conducted. After testing was concluded, a component inspection was conducted to assess the wear condition.

The concepts selected for detail design and test were:

- o Overlap valve
- o Flow augmentation/load recovery valve
- o Intelligent pump

The overlap valve, Figure 102, will reduce the valve null leakage. A detailed description of this concept is contained in Section 3.1.1. The parameters that were measured are:

- o No-load max rate with valve reversal
- o No-load frequency response
- o Null leakage (-65°F and 275°F)
- o Actuator hysteresis and threshold

Flow augmentation/load recovery valves, as shown in Figure 103, will reduce the pump flow demand when an actuator is operating at low load/no-load/assisting loads. A detailed description is given in Section 3.1.1. The parameters measured were:

- o No-load max rate with valve reversal
- o Variable load max rate (-65°F and 275°F)
- o No-load frequency response
- o 40 percent of resisting and assisting stall load frequency response

The intelligent pump, Figure 105, will maintain a minimum 3,000 psi system pressure and respond with higher pressures to 8,000 psi when required at actuators. A detailed description of this concept is shown in Section 3.1.1. The parameters measured were:

- o Pump hysteresis
- o Pump step response (no-load stabilator inputs)
- o Pump frequency response
- o Stabilator actuator no-load frequency response
- o Pump pulsation mapping at 3,000 and 8,000 psi

a. Failure Modes - Failure modes were defined and each concept was tested in the failed mode. The overlap valve has no failure modes peculiar to the concept and will be the same as the conventional servovalve. The flow augmentation/load recovery valves were tested for the following failure modes:

- o Jet pump check valve failed closed; perform no-load max rate test
- o Jet pump check valve failed open; perform no-load max rate test
- o Load recovery check valve failed closed; perform variable assisting load max rate test
- o Load recovery check valve failed open; perform variable assisting load max rate test

The intelligent pump was tested in the following failure mode:

- o Pump compensator failed to low pressure setting; perform stabilator actuator no-load frequency response and max rate tests

b. Data Requirements - The test system, Figure 3, was instrumented so that all critical parameters were recorded. The following overlap valve parameters were recorded:

- o Stabilator valve position
- o Stabilator actuator position/rate
- o Stabilator valve inlet and outlet pressures
- o System return flow
- o Various system temperatures
- o Stabilator input command
- o Pump input torque

The flow augmentation/load recovery valve data requirements are similar to those of the overlap valve concept. The following parameters were recorded:

- o Stabilator valve position
- o Stabilator actuator position/rate
- o Stabilator valve inlet and outlet pressures
- o System return flow
- o Various system temperatures
- o Pump input torque

The intelligent pump requires several additional parameters to be monitored because of the variable pressure characteristics. The following parameters were recorded:

- o Stabilator valve position
- o Stabilator actuator position/rate
- o Pump command
- o System return and pump case drain flow
- o Pump and stabilator valve pressures
- o Pump input torque
- o Various system temperatures

SECTION IV
PHASE III - DETAIL DESIGN

Phase III consisted of the following five tasks:

- 1) Final Performance Analysis and Design Definition
- 2) Specification Definition
- 3) Manufacturing Drawings of Pump and Actuator
- 4) Test Plans
- 5) Preliminary Hazard Analysis

4.1 TASK 1 - FINAL PERFORMANCE ANALYSIS AND DESIGN DEFINITION

4.1.1 System Selection and Definition - The definition of a test system was required to evaluate the low energy consumption concepts selected in Phase II of the LECHT program. The concepts tested were:

- o Variable pressure pump
- o Overlap valve
- o Flow augmentation
- o Load recovery valves

To evaluate the concepts, the demonstration system for the FWFRHS was modified to include a variable pressure pump and a loaded stabilator actuator with provisions for an overlap valve, flow augmentation devices and load recovery valves.

Figure 260 schematically represents the LECHT test setup. The system was made up of two main sections. A central system consisted of the variable pressure pump, a reservoir, filter, heat exchanger and appropriate relief valves. Check valves provided system pressure and fluid conditioning while a distribution system provided the means to transmit power from the central system to the remotely located actuator.

4.1.2 Variable Pressure Pump - Tailoring output pressure and flow characteristics of a hydraulic pump to the instantaneous and steady-state demand of the aircraft flight control system, offered significant benefits:

- o Pump pressure and flow was automatically reduced during flight conditions not requiring high control surface rates or loads, such as subsonic cruise. The average leakage flows were less than those for a constant high pressure (8,000 psi) system, resulting in lower system heat rejection. For example, reducing operating pressure from 8,000 to 3,000 psi could reduce the heat generation by 75 percent. The primary savings accrued from reduced pump heat rejection.
- o Engine power extraction was reduced, saving fuel and extending mission range.

Variable pressure (Intelligent) pumps for 3,000 psi systems were investigated extensively by MCAIR. The F-15 iron bird was used with "Smart Pumps" (one Abex, one Vickers) in each PC system to develop pump control

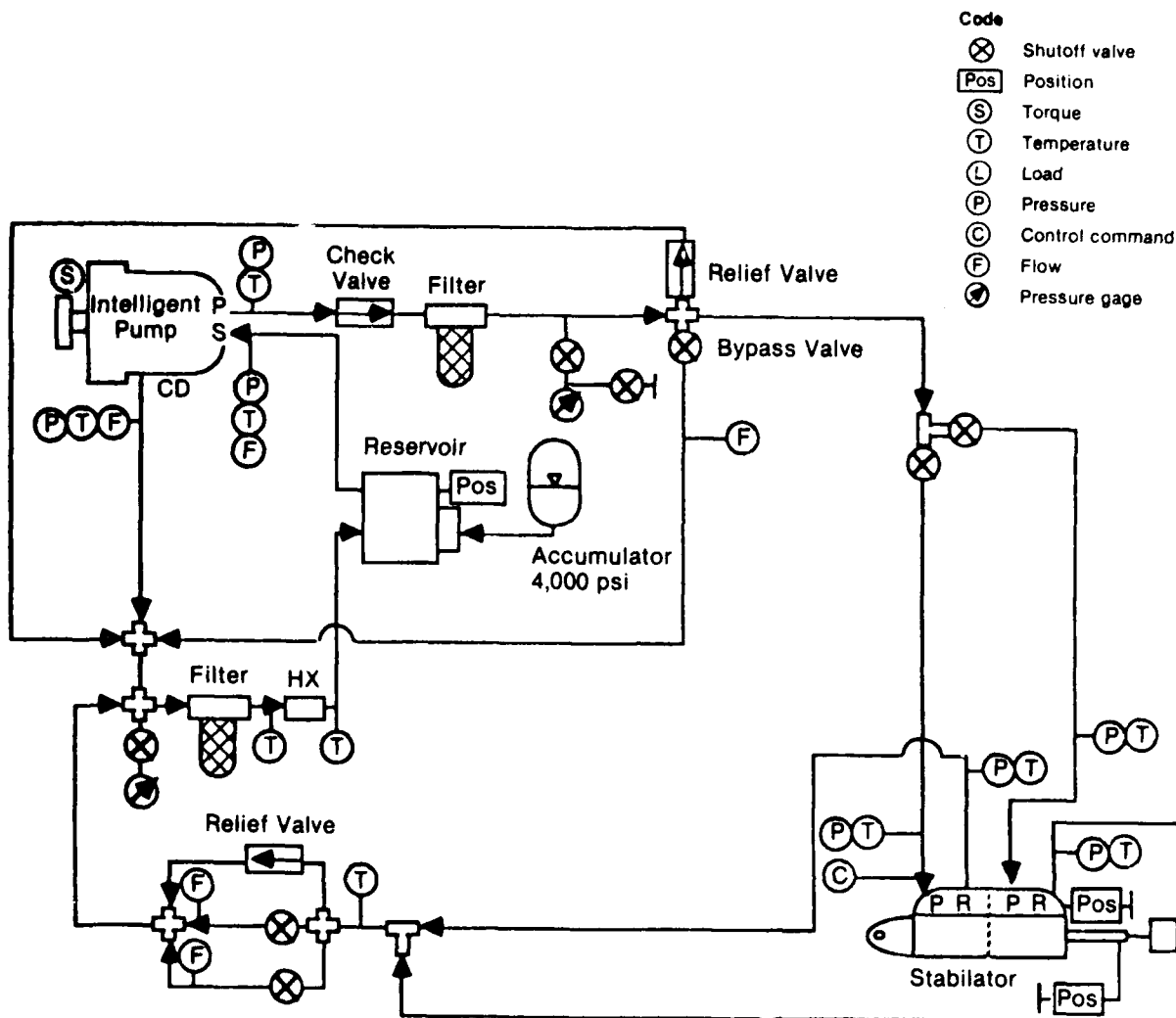


Figure 260
LECHT Demonstration System Schematic

concepts, including the effects of reduced control stability. The two PC pumps were then replaced by two Abex "Smart Pumps," and two Vickers Digital Hydraulic Control (DHC) pumps replaced the utility pumps. Figure 261 represents a simplified schematic of the Aircraft Power Controller (APC) used in these tests to schedule system pressure based on estimated surface power demand. The servovalve error signal of each flight control surface was a direct measure of the power demand of that surface. The error signal, available from the Flight Control Computer (FCC), was multiplied by the servovalve flow gain (G) to produce an estimated flow demand for that surface. It was then summed with the other surfaces to produce an overall flow demand (D) for the aircraft. The overall flow demand was fed into a computer to determine the system pressure necessary to accomplish the flight control command.

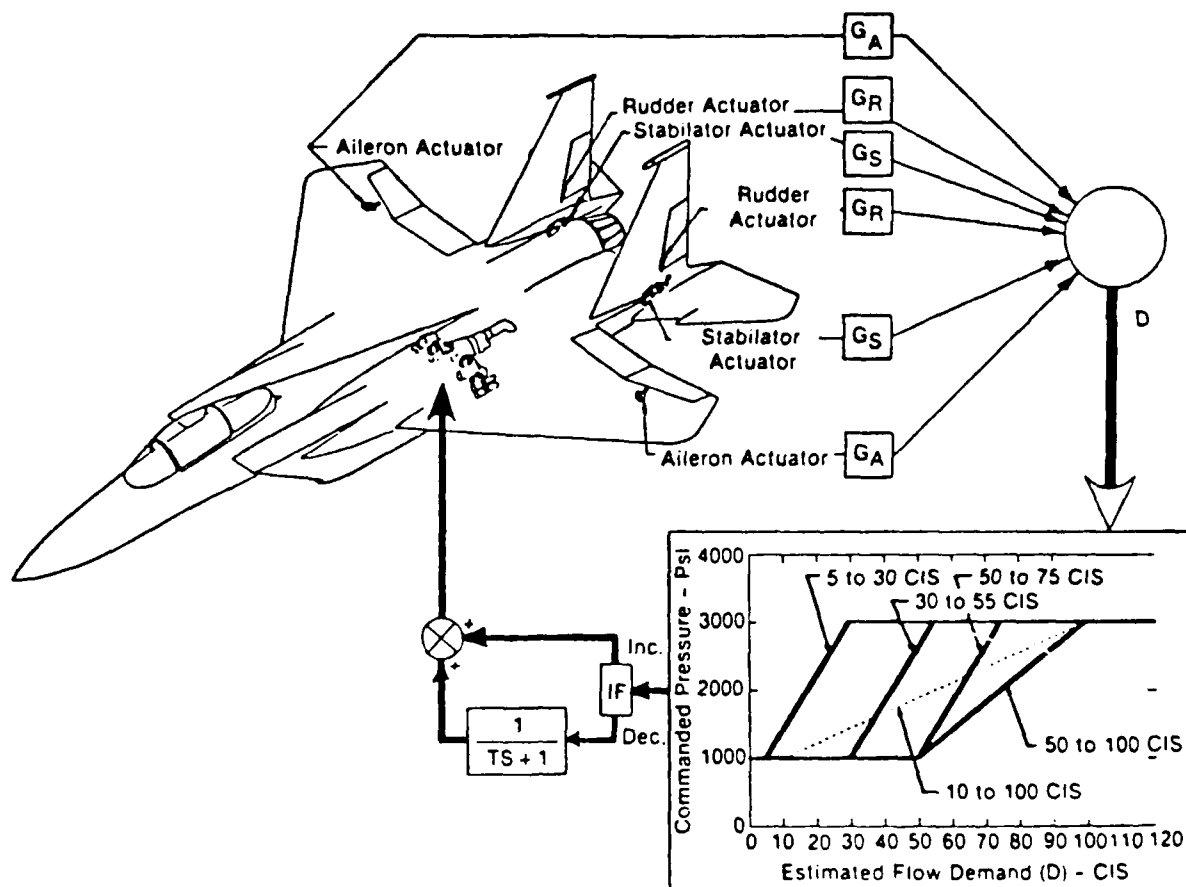


Figure 261
Aircraft Power Controller (APC) Schematic

The five pressure schedule curves shown in Figure 261 were used to investigate the effect each had on aircraft performance. Constant 1,000 and 3,000 psi pressure command curves were used to investigate worst case failure mode performance and established baseline aircraft performance respectively. The pressure command signal was delivered directly to each of the variable pressure pumps if the command pressure was increasing. For decreasing pressure, the signal was subjected to a two-second lag filter reducing the number of pressure cycles that adversely affected component fatigue life.

Acceptance tests, including hysteresis and linearity, frequency response and step response, were carried out on each pump. The closed loop, variable pressure pump, showed superior performance characteristics to the open loop design. The differences in performance were due to differences in design philosophies. The Abex unit was operated open loop and commanded to 3,000 psi at 0 volts, and 1,000 psi at 10 volts. An electrical failure commanded system pressure to 3,000 psi. The Vickers unit was operated closed loop and commanded to 0 psi at 0 volts, and 3,000 psi at 10 volts. An electrical failure closed a solenoid valve and switched control from the digital, variable pressure compensator to a conventional hydromechanical compensator set at 3,000 psi.

Five current fighter pilots "flew" the F-15 iron bird with variable pressure pump control. Each reported aircraft handling qualities for discrete maneuvers in pitch and roll. A blind test of the five APC configurations was performed. Overall, comments about the variable pressure pump control were very close to the baseline aircraft (constant 3,000 psi) for the 5 to 30 cis and the 30 to 55 slopes, which were preferred by the pilots.

The variable pressure pump performed as well as a constant pressure pump in providing the required hydraulic power to each flight control surface. The lack of availability of a Vickers 8,000 psi variable pressure pump, led to the selection of on Abex "Smart Pump" for the LECHT demonstration testing, despite the performance advantages of the closed loop design.

4.1.3 Overlap Valve Selection - A significant reduction in energy consumption at 8,000 psi system pressure, can be achieved by the control of servoactuator internal leakage. Evaluations of various fluid leakage characteristics vs. pressure and valve overlap, were accomplished under Independent Research and Development (IRAD) and a typical result for CTFE is presented in Figure 262. An overlap of 0.003 inch was selected for the LECHT

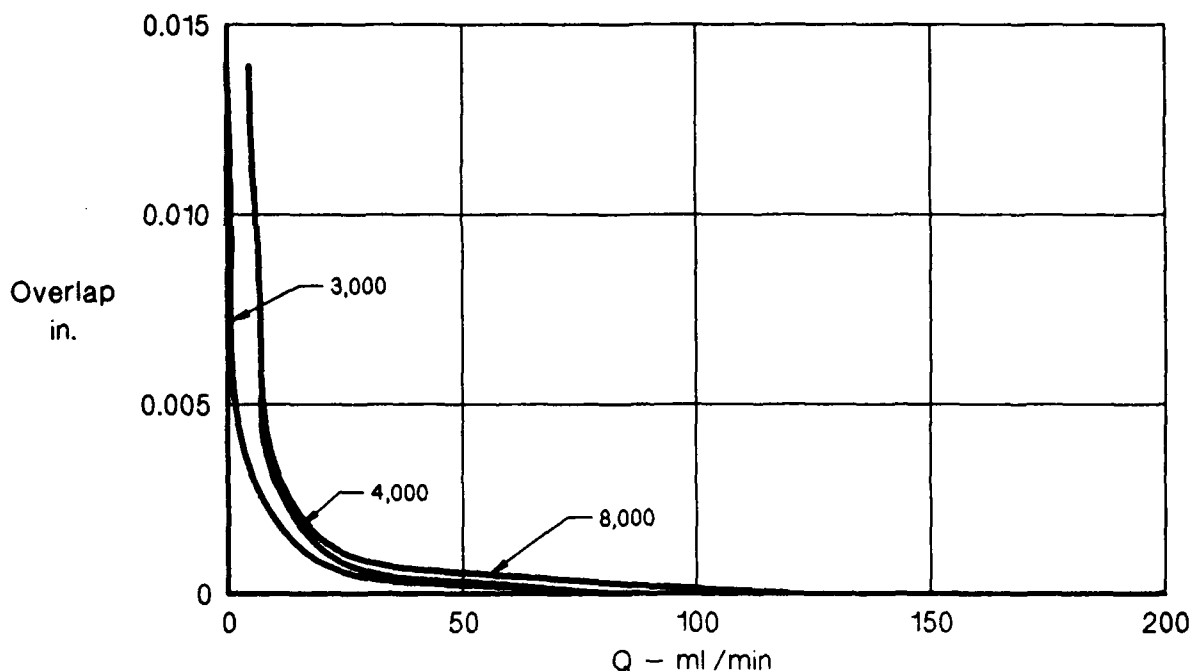


Figure 262
CTFE Fluid Characteristics

demonstration testing on the basis that very little reduction in leakage was gained by larger overlap. An order of magnitude reduction in null leakage can be accomplished with 0.003 inch and larger overlaps.

The benefits of such a reduction in null leakage are presented in Figure 263. The use of force motors and overlapped valves can cut system heat rejection by approximately two-thirds, (from 108.5 hp to 36.0 hp), at 8,000 psi.

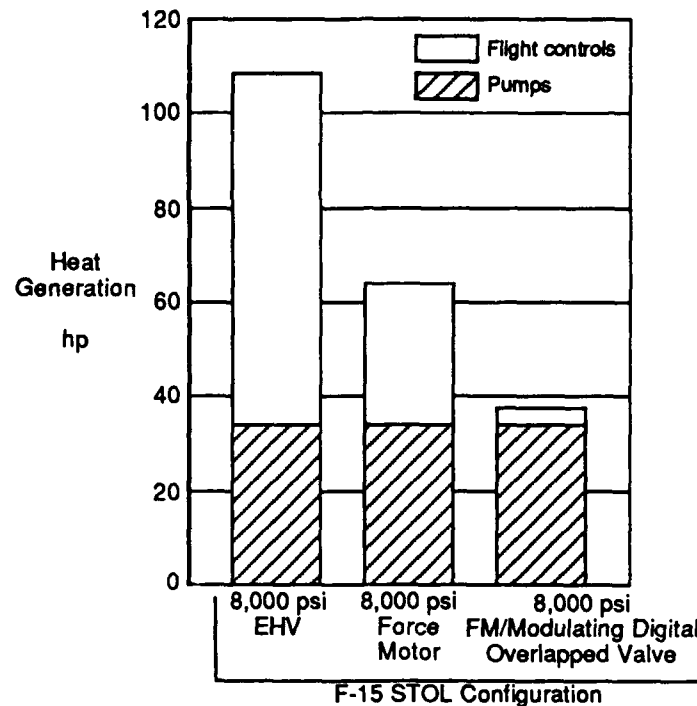


Figure 263
Heat Generation vs. System Variables

Another apparent benefit of the overlap valve was the elimination of the effects caused by signal noise. The switch from analog to digital computers, digital feedback circuitry, etc., resulted in the control of signal "noise". The "noise" can cause oscillations of the main control valve which resulted in a higher integrated null leakage, i.e., increased heat rejection. Overlap valves can eliminate this problem.

4.1.4 Flow Augmentation Devices - The flow augmentation device (jet pump) was first suggested by Parker Bertea approximately 20 years ago for the Boeing SST. The jet pump, (shown schematically in Figure 264), utilized the relatively high return pressure and flow leaving a servoactuator during high surface rate conditions, to supplement the inlet flow to that actuator. This cut pump flow demand by nearly 50 percent and reduced the weight, cost and heat rejection of all the central system components, i.e., the pump, filter manifold, heat exchanger, valves, tube assemblies, etc. The system incurred slight to moderate performance degradation, (in a noncritical area of the power transmission curve), and there was an increase in cost and complexity of the servovalve manifold.

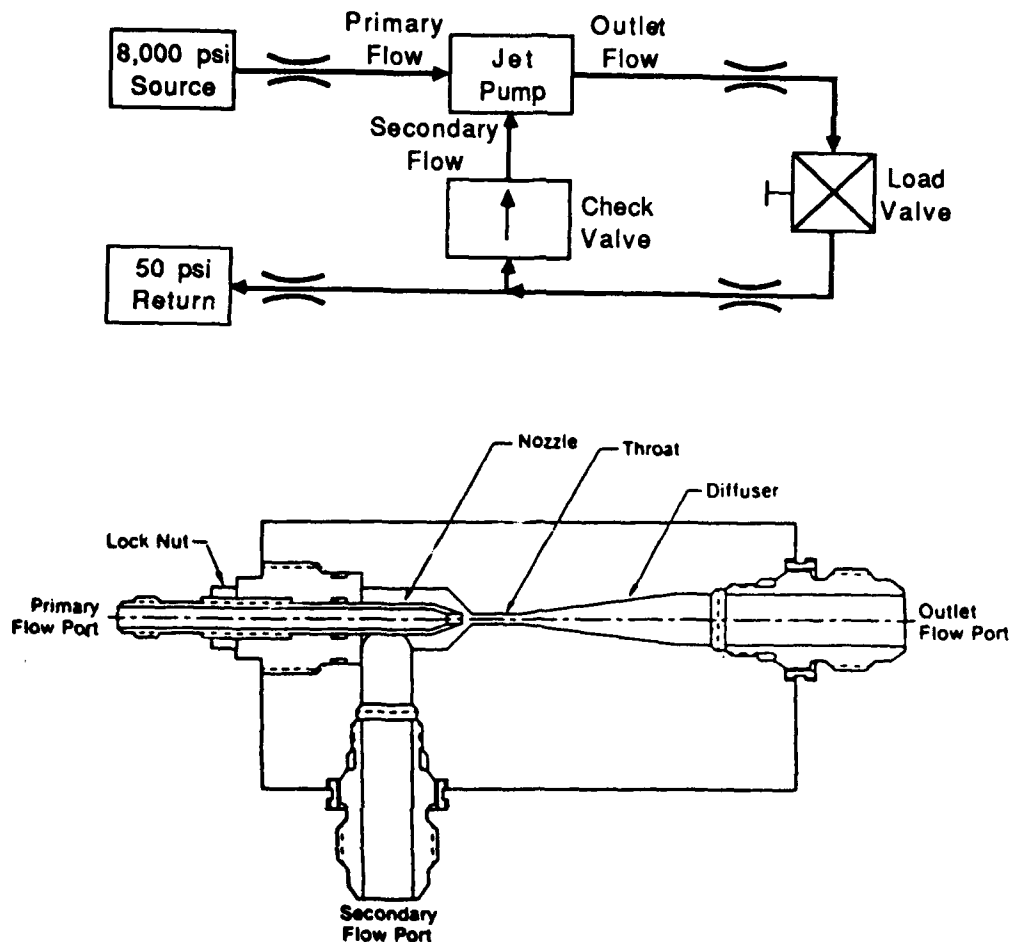


Figure 264
Jet Pump Schematic

Testing was conducted on a 3,000 psi MIL-H-83282 hydraulic bench. The linear position of the jet pump primary flow nozzle relative to the throat was adjusted for optimal performance. A position of twice the nozzle diameter from the end of the nozzle to the throat opening, showed the best results, but the performance varied only moderately over a range of positions. Some improvement in performance was noted when the surface finish of the throat and diffuser was improved. The length of the throat was also adjusted to determine the optimal length of mixing. A length of seven times the throat diameter was found to be the optimum.

Another factor investigated and probably the most crucial, was the internal geometry of the nozzle. The difficulty of accelerating the fluid from a large area through the small nozzle area with the least amount of friction loss, required that care be taken in determining the detail of the nozzle design. Three configurations were investigated, with the "wedge" nozzle chosen on the basis of machinability and equivalent performance to the most difficult design.

Flow augmentation devices were optimized for the LECHT stabilator servoactuator, sized for 8,000 psi, using the flow augmentation computer program developed by MCAIR. A comparison between flow augmentation and conventional systems is made in Figure 265. In the resisting load portion of the performance envelope shown, the crosshatched region is lost when flow augmentation is used. Aerodynamicists on the F-18, determined that this lost region would not adversely affect F-18 handling qualities, and F-18 simulator

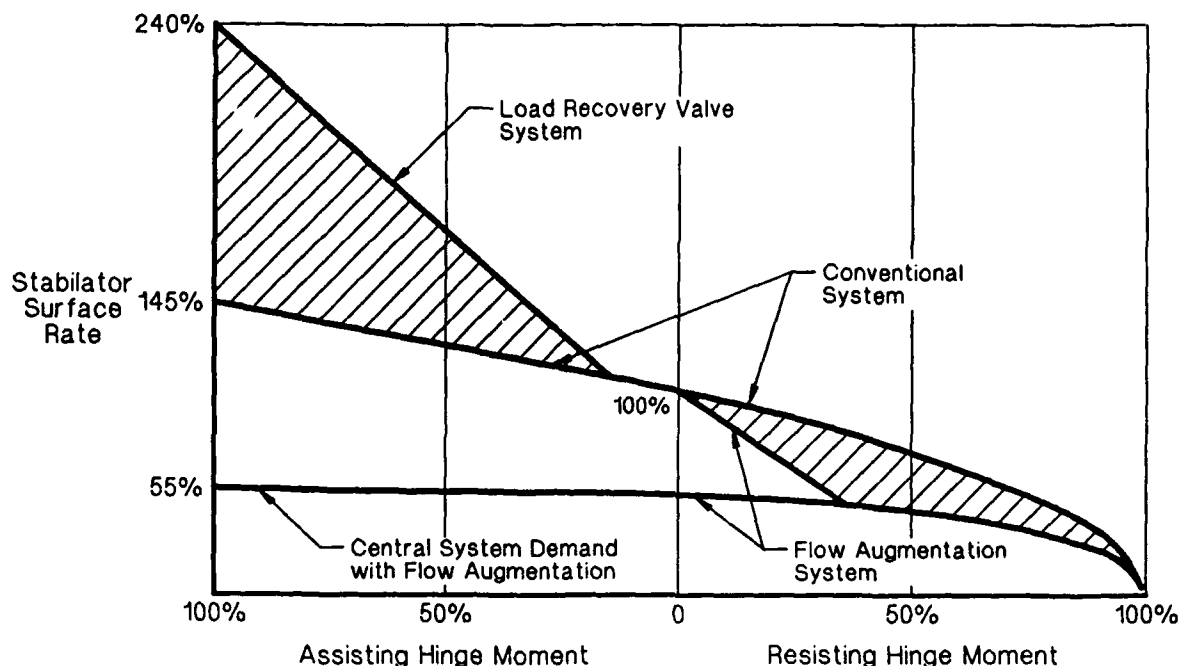


Figure 265
Stabilator Surface Rate vs. Hinge Moment Comparison of
Conventional vs. Flow Augmentation System

evaluation confirmed this. It is important to remember that the performance parabola representing conventional system performance presents the hydraulic systems capabilities and does not necessarily present aircraft performance requirements. In applying the flow augmentation concept, one must evaluate the specific aircraft requirements to verify applicability.

4.1.5 Load Recovery Valve Design - The highest power consumption by a flight control hydraulic system occurs during periods of sustained aircraft maneuvering, such as takeoff, landing and air combat. In present day hydraulic systems, this activity can force the system pumps to run at maximum capacity, extracting the maximum possible energy from the power plants.

Actuator activity during these flight phases consists of repeated cycling about a given position. The actuator faces a resisting load as it pushes the surface into the airstream, then an aiding load as it withdraws the surface. Present systems are designed as if the actuators were presented

with a resisting load continuously throughout this cycle. Load recovery valves offer a means of reducing hydraulic power levels during aided load conditions. They allow a 2, 3, or 4 times (as desired) increase in assisting load rates without any additional demand on the central system pump.

As shown in Figure 266, the load recovery valve concepts are readily applicable to linear hydraulic actuators, presenting a way of interconnecting the actuator chambers during aided load conditions. The system is identical to existing anticavitation systems on many actuators, though the check valves may be appreciably larger. Under aiding loads, the flow is supplied to the cavitating chamber of the actuator from the return side of the system as well as the pressure side. By essentially short circuiting the actuator chambers during aided loads, the concept offers the potential of increasing the actuator rate while decreasing flow in the central system. Lower flow requirements during this portion of the cycle, effectively reduces the total power consumption of the hydraulic system.

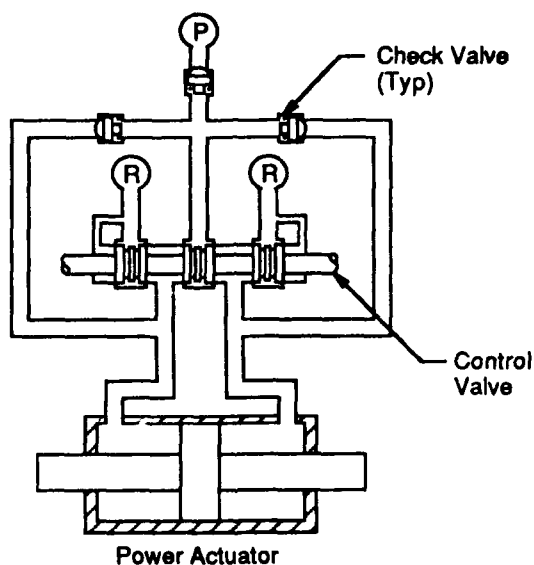


Figure 266
Schematic of Load Recovery Valves

Figure 265 shows the change in performance in the assisting load portion of the performance envelope associated with the load recovery valve concept. The crosshatched triangle represents increased rate available for the given LRV design, but this can be modified higher or lower by changing the design. The load recovery valves and flow augmentation devices are complementary. The LRV prevents cavitation of the flow augmentation concept when jet pump performance is optimized for the resisting load portion of the performance envelope.

4.1.6 FAST Actuator - The Flow Augmented Servovalve Technology (FAST) actuator includes both flow augmentation devices and the load recovery valves, mounted integrally in the valve manifold. A FAST actuator was used for verification of the previously discussed concepts and the results are presented in Section 5.2.1.b.

The combination of concepts can result in asymmetric rates for complementary control surfaces, such as the two separate stabilators on the F-15 and F-18. Consider a rolling pull up where one surface may maintain a resisting load and the other surface may maintain an assisting load. Reviewing Figure 265 again, a significant rate difference can exist compared to "pull up" or "push over" maneuvers where load direction and magnitude are approximately the same for both surfaces. The asymmetric condition associated with using both concepts simultaneously was flown on the F-18 simulator, with no adverse affects noted by pilots. However, each aircraft must be evaluated independently in order to verify compatibility or the need for modification to the system.

The Figure 265 classic rate vs. hinge moment plot, is convenient for presenting and discussing performance. The curves represent the locus of an infinite number of rates given a constant hinge moment. Since all testing was done using "real world" varying load, the test results must be presented differently. The conventional system and the FAST actuator concept performances are compared and evaluated to verify or deny the concept's capability in Section 5.2.1.b.

The 8,000 psi stabilator actuator was loaded approximately as it would perform in the aircraft, as shown in Figure 267. Approximately two-thirds of the travel was associated with getting the surface leading edge down for aircraft pull ups/rolls and the other one-third with getting the leading edge up for push overs/rolls. Therefore, the peak load was associated with maximum leading edge down surface position. The maximum leading edge up load was approximately 48 percent of the leading edge down maximum load.

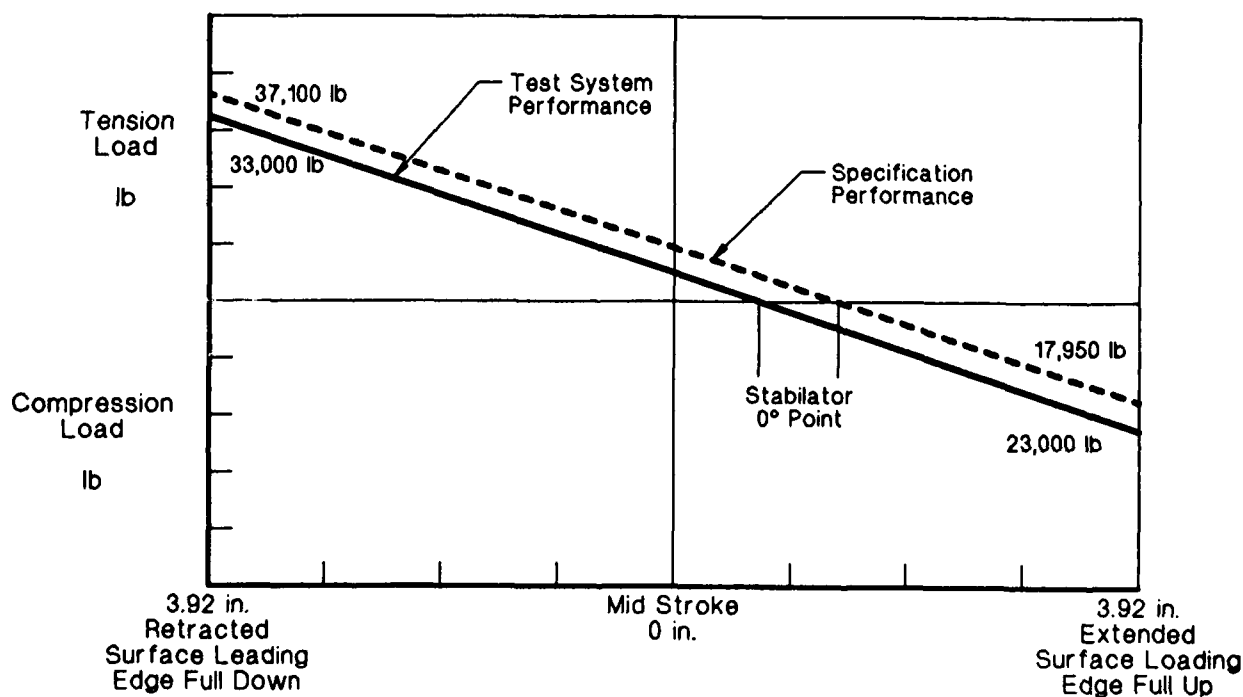


Figure 267
Stabilator Actuator Loading

4.1.7 Distribution System Losses - Sizing of the demonstration system was approached with the intent of evolving a minimum weight hydraulic system with acceptable performance characteristics. Since Phase I studies identified the distribution system as a prime benefactor of the volume/weight reduction possible with higher system operating pressures, use of minimum size tubing was made a general requirement. Specification of a low pressure drop actuator control valve, allowed the usage of small diameter tubing to control actuator velocities. Such tubing, in conjunction with CTFE fluid density, presented the possibility of very high water hammer transient pressures. Control of such transient pressures, was achieved by using Local Velocity Reduction (IVR), which uses a larger pressure line immediately upstream of the actuator to reduce fluid velocities, and therefore water hammers.

For LECHT, the system had to be adaptable to actuators with and without jet pumps. The optimized design, used when the jet pumps were not in the system, was to size them for a stall load and no-load rate, and to use a parabolic power curve to determine power transmission. The distribution line losses at no-load flow, see Figure 268, are as follows:

- o Pressure side - 5,800 psi
- o Actuator/valve - 800 psi
- o Return side - 1,200 psi

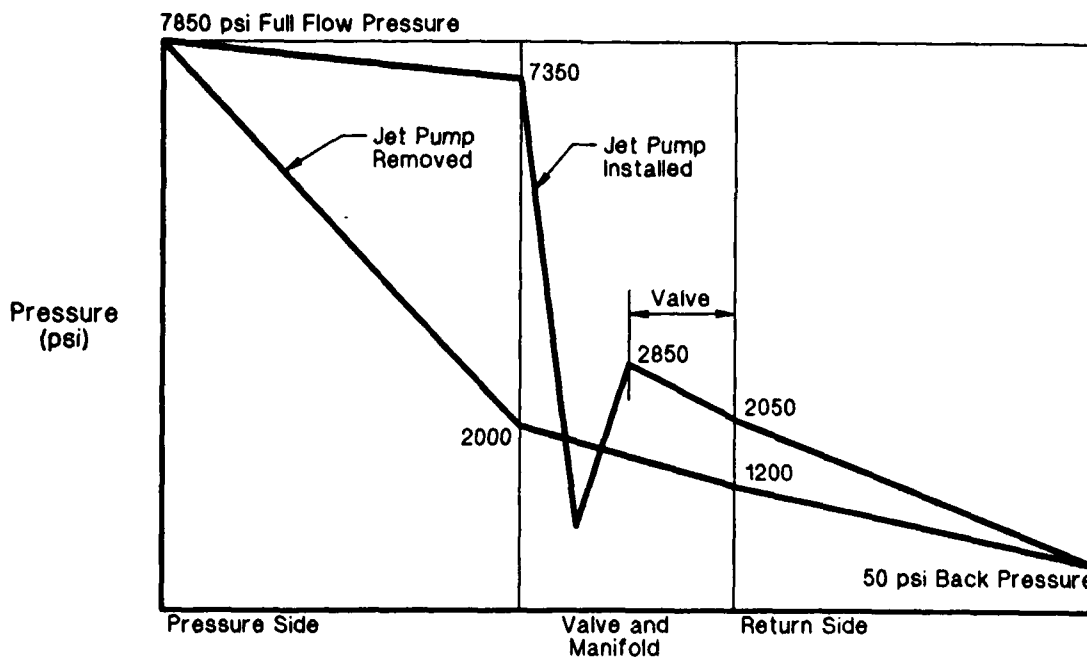


Figure 268
Pressure Loss Distribution

These pressures were based on having 7,850 psi at full flow, with a reservoir pressure of 50 psi. Allotting most of the pressure drop to the pressure side distribution system, much smaller tubing can be used, thereby saving system weight. On the negative side, the smaller line sizes cause very high fluid velocities, which in turn can cause excessive water hammer transients. To control these transients, the concept of LVR was utilized.

For a system with flow augmentation devices installed, the system was also sized for a stall load and no-load rate. Because of the basic inefficiencies of the jet pumps, significant pressure needs to be available at the actuator, see Figure 268. The distribution line losses used for the LECHT test with jet pumps were as follows:

- o Pressure side - 500 psi
- o Actuator/valve - What Augmentor and Valve Required
- o Return side - 2,000 psi

Despite the fact that the distribution system dropped only 2,500 psi instead of 7,000 psi, the system weight change was not significant. The pressure drop on the pressure side was 500 psi, (vs. 5,800 psi), but approximately half the flow was required because of flow augmentation. This yielded a pressure side distribution system somewhat heavier than one designed for the actuator without a jet pump. However, this was partially offset by the small weight reduction in the return side distribution system, which had an increased pressure drop of 2,000 psi, (vs. 1,200 psi), at half the flow.

To eliminate the need to change transmission lines every time the jet pump was installed in or removed from the actuator, the pressure drop through the demonstration system was controlled by a pair of needle valves on the pressure side, and a single needle valve on the return side of each system. Therefore, when the jet pumps were removed from the actuator, the needle valves were closed on the pressure side to provide the necessary 5,800 psi drop. When the jet pumps were added, the needle valves on the pressure side were opened wide, while the return side needle valve was partially closed.

To verify the above concepts, the computer simulation programs SSFAN, and HYTRAN were used where applicable. SSFAN was helpful in sizing the tubing for no-load rates and stall loads, while HYTRAN was used to verify the concept of local velocity reduction.

4.2 TASK 2 - SPECIFICATION DEFINITION

4.2.1 Pump - The Abex Corporation converted two 8,000 psi, 15 gpm pumps, (model AP6VHP-3), to variable pressure, (3,000 to 8,000 psi), intelligent pumps for use in MCAIR's 8,000 psi efforts.

Performance parameters were defined in Procurement Specification CTFE-1, which was released for the FWFRHS program. The pump was required to operate on CTFE fluid filtered to five-micron absolute. A proof pressure of 12,000 psi and a burst pressure of 20,000 psi at the outlet port was specified. The pump must be capable of immersion in CTFE fluid at 275°F for 72 hours, and

the equipment must be capable of starting and operating at -65°F inlet fluid temperature. A rated delivery flow of 15.0 gpm at a minimum full flow pressure of 7,850 psi was required, and a rated speed was specified at 4,000 rpm. A minimum inlet fluid pressure was set at 40 psia, with an inlet fluid temperature of 215°F. The maximum displacement was set at 0.95 in³/rev. Peak transient pressures were specified as 800 psi maximum above full flow pressure for transition from 100 percent flow to zero flow, and a minimum of 2,000 psi below rated discharge pressure for transition from zero to 90 percent flow.

The pump control used a standard Abex 410 electrohydraulic valve (EHV) modified for special feedback spring configuration, with provisions for higher return pressures and a dropping orifice to limit control pressure to 3,000 psi at 8,000 psi supply, and 1,000 psi at 3,000 psi supply. The controller was designed with 0 volts at 8,000 psi and 10 volts at 3,000 psi, which allowed it to fail at 8,000 psi. Two pumps were procured with MCAIR funding, one of which was dedicated to the LECHT demonstration and endurance testing, while the other served as a backup.

4.2.2 Actuator - Parker Berteau was contracted to design and fabricate one 8,000 psi, CTFE version of the F-15 stabilator servoactuator to the requirements documented in procurement specification CTFE-2. The actuator was a fixed body, dual tandem hydraulic servoactuator with the same force output, stroke and mechanical mounting attachments required for the F-15. Specifically, the actuator was required to provide 37,100 ± 600 pounds of tension and 42,900 ± 600 pounds of compression with normal operating supply, (8,000 psi), and return, (50 psi), pressures with and both systems operating. Stroke was specified to be 7.70 inches. The actuator differed from the F-15 in that there was no mechanical main ram feedback, Control Augmentation System (CAS) lock, CAS shutoff valves, or a differential pressure sensor. The tandem master control valve was a two-stage design controlled by a spring centered electromechanical force motor located within the valve manifold. The main ram position and valve position loops were closed by outside external electronics which utilized signals from the valve and main ram LVDT position transducers. In addition, provisions for hole-in-the-wall stiffness testing were also specified.

The tandem valve was required to operate normally with either or both hydraulic systems pressurized. The second stage valve was sized to provide a main ram velocity of 8.2 in/sec, with a total valve pressure drop equal to 10 percent of system pressure with the valve full open. Two main control valves were provided, one with 0.003-inch overlap, and the other line-to-line. The valve manifold was required to be a flight weight, flight worthy design, with provisions for a single-stage or two-stage direct drive valve, load recovery check valves and flow augmentation devices. Parker Berteau provided the load recovery valves, which are essentially enlarged anti-cavitation check valves, although sizing of the LRVs was accomplished at MCAIR.

Cavities were required in the manifold for installation of the MCAIR provided flow augmentation devices. The jet pump hardware for the 8,000 psi stabilator servoactuator was designed and optimized for lightweight installation in the stabilator servovalve manifold. The Lee Company was contracted to fabricate the nozzle and diffuser shown in Figure 269. C. E. Conover and Company supplied modified EPDM (material XD268) seals and Revonoc 18158

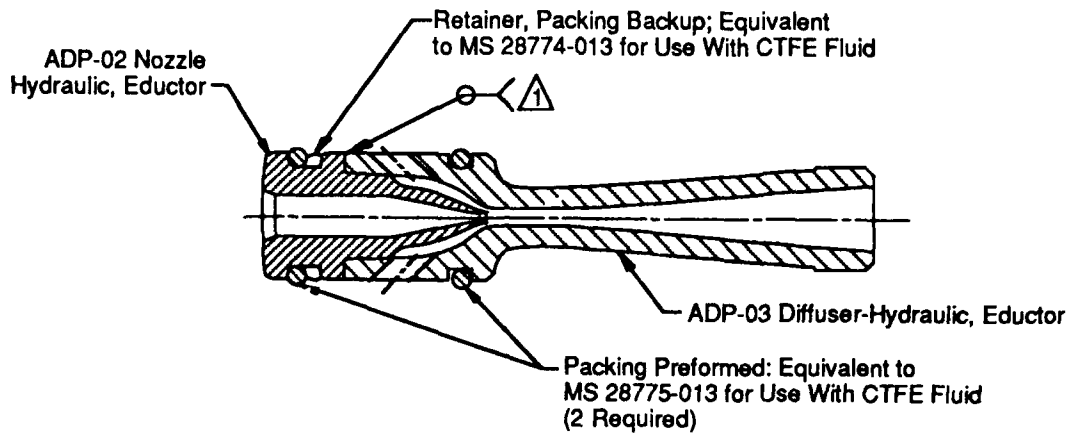


Figure 269
8,000 psi Stabilator Flow Augmentation Device

back-up rings for CTFE fluid compatibility between -65°F and 275°F. Two test manifolds were fabricated by the MCAIR development laboratory to house the 8,000 psi stabilator flow augmentation devices and 8,000 psi low loss check valves supplied by Parker Berteau. MCAIR also worked with Crissair Inc., to design and fabricate a -10 Rosan fitting, (5/8 inch), which incorporated an 8,000 psi check valve and 225-micron filter with only 30 psid pressure drop at 3 gpm flow (CTFE fluid). The check valve fitting was used upstream of the flow augmentation device and significantly miniaturized the total flow augmentation package.

The actuator was required to meet the F-15 SMTD aircraft frequency response characteristics over a temperature range of +20°F to 275°F, with some degradation of performance at temperatures below +20°F. The unit was specified to provide a zero load, open loop, (main control valve hard over), main ram velocity of 8.2 ± 0.8 inches per second with normal supply and return pressures.

Berteau was required to prepare and execute an acceptance test procedure to assess the performance and design of the actuator, using CTFE nonflammable fluid, and including proof pressure, functional and leakage tests. The test procedure was approved by MCAIR and AFWAL/POOS prior to testing.

4.2.3 Miscellaneous Component Suppliers and Description - The remaining demonstration system components were retained from the FWFRHS test, with the exception of the reservoir. The old F-4 power control system 3,000 psi bootstrap reservoir, was replaced by an 8,000 psi Metal Bellows Corporation (MBC) design.

a. Filtration - The filter assemblies were retained from the FWFRHS demonstration system. Aircraft Porous Media Inc. (APM), supplied the filter units for the demonstration system. The units had button-type differential pressure indicators to signal the excessive pressure drop a clogged element would produce. The return filter incorporated a bypass valve to prevent pump starvation which could occur during periods of transient high flow through a partially contaminated element.

The return filter, (APM Assembly No. AD-3258-16Y73), was a 3,000 psi design. APM was familiar with CTFE fluid properties, because they had supplied similar filters for Boeing Military Aircraft Company's evaluation of Halocarbon CTFE fluid. Filter elements for this unit met MIL-F-8815C five-micron absolute particle removal standards.

The pressure filter (APM Assembly No. Ad-4097-16Y69) utilized a housing for a commercial unit APM manufactured for use in high pressure test stands and other nonairborne applications. This housing, rated for 10,000 psi operating pressures, was of stainless steel construction. The high fluid velocities attendant with high pressure hydraulic systems, led APM to recommend three-micron absolute filtration for this unit. This recommendation was adopted, and filter elements meeting MIL-F-81836 requirements for three-micron absolute filtration were used. Figure 270 summarizes the filter data.

	Pressure Filter	Return Filter
Rated Pressure (psi)	8,000	3,000
Proof Pressure (psi)	12,000	4,500
Burst Pressure (psi)	20,000	7,500
Rated Flow (CTFE Fluid) (gpm)	22	22
Pressure Drop (psi) (Clean Element, Total Assembly 100°F)	30	32
Maximum Particle Passed (micron dia)	3	5
Temperature Range (°F)	- 65 to 275	- 65 to 275
ΔP Indicator Activation (psid)	80	70
Bypass Valve Cracking Pressure (psid)	—	100

Figure 270
Demonstration System Filter Data

b. Reservoir/Accumulator - An 8,000 psi bootstrap reservoir, manufactured by MBC, was selected as the demonstration system reservoir. The reservoir was a straightforward bootstrap design, with approximately 160 cubic inches maximum capacity. A bootstrap pressure of 4,000 psi provided a reservoir pressure of 50 psi. The bootstrap pressure was provided by a 4,000 psi accumulator. A manual bleed port allowed removal of air which might have collected in the top of the reservoir. The accumulator, also manufactured by MBC, was an 8,000 psi design, with a 50 cubic inch capacity and a nitrogen precharge pressure of 4,000 psi.

c. Heat Exchanger - The heat exchanger requirement for the demonstration system was met by utilizing an oil to water heat exchanger previously used in the FWFRHS testing. An Ametek Company type AHT-2-A-SS shell/tube heat exchanger rated for 15 gpm on the shell (hydraulic) side and 25 gpm on the tube (water) side was utilized. All fluid from the pump case drain and actuator return circuits was routed through this heat exchanger before being returned to the reservoir and pump suction lines. Rated heat transfer of the unit was 6,566 BTU/min and hydraulic fluid temperature was controlled by modulating the cooling water flow. Design pressure was 350 psi for the hydraulic side and 200 psi for the water side. The design allowed for a single pass through the unit for the hydraulic oil and two passes for the water. Dimensions of the shell side were quite large, so there was very little pressure drop through the unit, even at maximum flows.

d. Check Valves - System check valves were retained from the FWFRHS demonstration system and were used to prevent backflow in the pump outlet and case drain circuits. These were commercially available devices manufactured by Circle Seal Corporation. Made of 17-4 stainless steel and rated for 10,000 psi, the check valves had a nominal cracking pressure of 5 psid and utilized soft (O ring) poppet/seat seals. Both check valves had a -6 tube size fitting at each end.

e. Relief Valves - High pressure relief valves were retained from the FWFRHS demonstration system, which were existing designs manufactured by the Circle Seal Corporation. The relief valve was an adjustable cracking pressure, single-stage relief valve with a metal-to-metal poppet/seat seal. It was set to relieve at 9,000 psi and was capable of a 10,500 psi full flow pressure. Adjustment of the cracking pressure was accomplished by subjecting the unit to the desired pressure on the hydraulic test bench, then reducing the spring preload until fluid began to flow from the outlet port. Both ports on the unit were -8 tube size, and the relief orifice diameter was 0.125 inches. CTFE fluid compatible PNF seals were installed prior to system usage.

4.3 TASK 3 - MANUFACTURING DRAWINGS

Envelope and assembly drawings of the Abex 15 gpm variable pressure intelligent pump, the Parker Berteau stabilator servoactuator and the MCAIR designed flow augmentation device (jet pump) are shown in Sections 4.2 and 5.1.1.

4.4 TASK 4 - TEST PLANS

4.4.1 Objective - The objective of this test was to evaluate the performance of low energy consumption concepts, and quantify energy savings compared to conventional designs. System design pressure was 8,000 psi and CTFE fluid was utilized. Performance tests evaluated individual concepts/components, and demonstrated the effects of failure modes. A 200-hour durability test was to follow performance testing.

4.4.2 Testing - The testing began with individual component/concept performance tests. Each concept's performance was compared to a baseline configuration without the concept. Failure modes were demonstrated where

applicable. A thorough system performance test was conducted with all the concepts installed to ensure that system requirements were met. Durability cycling was conducted at various load conditions for the proposed 200-hour test duration. Pump input torque was measured for energy comparisons with and without concepts active in the components. Other energy consumption tests, such as valve null leakage, were also conducted.

a. Performance Tests

(1) Pump Acceptance Tests - The following tests were performed on the Abex pump operating at a fixed pressure of 8,000, 5,500 and 3,000 psi to determine pump characteristics:

- o Physical Defects Inspection
- o Pump Leakage/Heat Rejection Tests
- o Pump Rated Flow
- o Pump Hysteresis and Linearity
- o Pump Frequency Response

(2) Baseline Configuration/Stabilator Acceptance Tests - The baseline configuration was used to determine the servoactuator performance characteristics. This configuration consisted of the pump operating at a fixed pressure, (3,000, 5,500 or 8,000 psig), a line-to-line valve installed, the load recovery valves replaced by a plug and the flow augmentation device removed.

The distribution system was adjusted to provide a 6,000 psi drop through the pressure side, (when operating at 8,000 psi), and to provide an actuator rate of at least 8.2 in/sec. The following tests were performed:

The following tests were performed:

- o Leakage Tests
- o Hysteresis
- o Threshold
- o No Load Frequency Response
- o No Load Step Response (Valve Reversals)
- o Loaded Step Response
- o 40 percent Stall Load Frequency Response

(3) Overlapped Valve Concept Tests - The following tests were used to evaluate the overlapped valve concept. The line-to-line valve was replaced by a 0.003-inch overlapped valve for these tests.

Null leakage data was used for comparison with the line-to-line control valve. The remaining performance tests were used to determine if there was any performance degradation with the overlap valve:

- o Null Leakage
- o Hysteresis
- o Threshold
- o No Load Frequency Response
- o No Load Step Response (Valve Reversals)
- o Loaded Step Response
- o 40 percent Stall Load Frequency Response

(4) Load Recovery Valve Concept Test - To test the load recovery valve concept, the four anticavitation check valves (load recovery valves), were installed in the actuator, replacing the plugs that were used for the previous tests. A loaded step response test was performed, to allow a comparison of energy consumption with the overlapped valve concept and the baseline actuator.

(5) Flow Augmentation Concept Tests - For the following tests, the flow augmentation device (jet pump), was installed, and the distribution system was adjusted to optimize the performance (low loss on the pressure side and higher loss through the jet pump). The loaded step response torque data will be compared to LRV test results.

- o No Load Step Response (Valve Reversals)
- o Loaded Step Response
- o No Load Frequency Response
- o 40 percent Stall Load Frequency Response

(6) Variable Pressure Pump Concept Tests - To test the concept of the intelligent pump, the pump was operated in the variable pressure mode for the following tests:

- o No Load Frequency Response
- o 40 percent Stall Load Frequency Response

b. Endurance Test - A 200-hour durability test was proposed in four blocks of 50 hours each, using the full up configuration, including the overlap valve, load recovery valves, jet pumps and operating with variable pressure. Before and after each block, a pump leakage test, a stabilator leakage test and a stabilator frequency response test were conducted. In addition, fluid samples were collected before and after each block and at the time of any failures. Eight ounce fluid samples were sent to AFWAL/MLBT. Before and after each block, one 72-second cycle at constant 8,000 psig system pressure, and another 72-second cycle with 3,000 to 8,000 psig varying pressure (using the controller settings selected in 4.4.2.a.(6)) was run to measure pump torque during the cycles. Figure 271 describes each block.

Load Block	Elapsed Time (hr)	Actuator at Full Retract (%)
1	50	100
2	50	100
3	50	100
4	50	100

Figure 271
Stabilator 50-Hour Test Blocks

The following table defines the stabilator servocylinder stroke cycle sequence used for the durability test. Each stroke was to be centered about the zero degree stabilator position. The stabilator actuator was cycled 100 times each 72 seconds at the stroke and cycling rates shown below in Figure 272.

Time (sec)	Stroke (%)	Cycling Rate (Hz)	Number of Cycles
0 - 12	2	2.5	25
12 - 24	30	0.5	5
24 - 36	2	2.5	25
36 - 39	100	0.4	1
39 - 42	10	1.0	2
42 - 48	70	0.5	3
48 - 60	2	2.5	28
60 - 72	10	1.0	11

Figure 272
Stabilator Duty Cycle
for 200-Hour Endurance Test

Fluid temperature was not to be controlled except that the pump outlet temperature was not allowed to rise above 200°F. Environmental temperature was room ambient.

Torque data recorded before and after each block was used to compare energy consumption per the 72-second cycle, between constant (8,000 psi), and variable (3,000 to 8,000 psi), pressure operation over the proposed 200-hour test.

4.5 TASK 5 - HAZARD ANALYSIS

The objective of the Preliminary Hazard Analysis (PHA) was to assess the hazardous risks associated with the LECHT concepts in an aircraft application, and as they are incorporated in the 200-hour durability test system. The test system, as shown in Figure 260, incorporated the following low energy consumption concepts:

- o Intelligent Pump
 - 15 gpm
 - 3,000 to 8,000 psi
- o Stabilator Actuator/Valve
 - flow augmentation/load recovery valve (see Figure 273)
 - overlapped spool

The study was performed to determine if the concepts would present any hazardous conditions. Also studied, was the effect that a component failure would have on system operation and life degradation. Initial safety

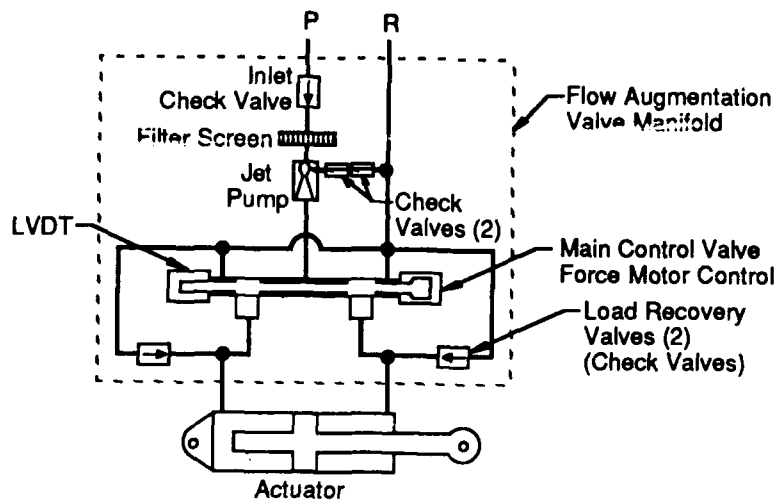


Figure 273
Flow Augmentation/Load Recovery Valve
(FA/LRV) Schematic

considerations were established to minimize the adverse effects of the concepts. The PHA was performed in accordance with MIL-STD-882A. A matrix format was used to document the analysis. A USAF approved form, MAC form 3413D, was utilized.

The first step in performing the PHA was to define and describe potential hazardous situations unique to each concept. Taken into consideration, were the environmental factors, human errors and material failures. To determine specific areas of hazardous condition concern, an AFSC DH 1-X safety design note checklist was utilized.

Next, the hazardous effects were determined. This defined what would physically occur relative to each failure etc., such as personnel injury or damage to equipment. Each hazard was then assigned a hazardous classification based upon the end effect of each hazardous condition. The criteria for classification is listed below:

Category I - CATASTROPHIC (1)

- o Will cause death or severe injury to personnel or system loss

Category II - CRITICAL (2)

- o Will cause personnel injury or major system damage, or will require immediate corrective action for personnel or system survival

Category III - MARGINAL (3)

- o Can be counteracted or controlled without injury to personnel or major system damage

Category IV - NEGLIGIBLE (4)

- o Will not result in personnel injury or system damage

A qualitative probability of occurrence was identified for each hazard. There are five levels ranging from frequent to extremely improbable. Listed below are the levels and rationale for choosing each:

Level

- o (A) FREQUENT: Likely to occur frequently
- o (B) REASONABLY PROBABLE: Will occur several times in the life of an individual item
- o (C) OCCASIONAL: Unlikely to occur in the life of one specific item
- o (D) REMOTE: So improbable that it can be assumed that this item will not experience
- o (E) EXTREMELY IMPROBABLE: Probability of occurrence cannot be distinguished from zero
- o (F) IMPOSSIBLE: Physically impossible to occur

A Real Hazard Index (RHI) was then assessed to provide a relative ranking of all identified hazards. The RHI is a dimensionless value determined by multiplying the Hazard Class by the Probability Level. Shown in Figure 274 is the RHI in a matrix format. The index numbers are a factor of both hazard category and hazard probability.

Figure 275 shows the Risk Assessment Criteria for each RHI. These criteria can be used as a basis for determining the amount of risk that a particular system will be subjected to.

Further analysis was performed to assess the method or means of identifying the hazard. Also, corrective action criteria was established to eliminate, reduce, or control the identified hazard. Finally, the recommended action and additional comments that should result from the PHA were

Hazard Category	Catastrophic I	Critical II	Marginal III	Negligible IV
Hazard Probability				
Frequent (A)	1A	2A	3A	4A
Probable (B)	1B	2B	3B	4B
Occasional (C)	1C	2C	3C	4C
Remote (D)	1D	2D	3D	4D
Extremely (E) Improbable	1E	2E	3E	4E

Figure 274
The Real Hazard Index

Real Hazard Index	Action
1A, 1B, 1C, 2A, 2B	Unacceptable Risk. Mandatory Correction, Elimination or Control. Requires Customer Acceptance if Not Corrected.
1D, 2C, 3A	Undesirable Risk. Attempt Should Be Made to Eliminate or Control. Requires MCAIR Approval for Risk Acceptance and Concurrence by Customer.
1E, 2D, 3B, 3C	Acceptable Risk. Customer Awareness Recommended.
2E, 3D, 3E, 4A 4B, 4C, 4D, 4E	Acceptable Risk

Figure 275
Rationale for Choosing RHI Level

established. This would recommend any changes or modifications necessary to minimize the hazardous affects of the LECHT concepts. The results and recommendations of the PHA are shown in Figure 276. According to this PHA, all identified hazards are an acceptable risk.

Component/ Function/Task	Fault/Hazard Description	Hazard Effects	Phase of Operation	Hazard Classification	Probability Level	Risk Hazard Index	(A) Method of Detection (B) Corrective Action	Recommended Action	Remarks
1.0 Intelligent Pump	1.1 Compensator Failure Pump Reverts to High Pressure Setting.	Excessive Heat Generation/ Possible Seal Degradation and Component Wear. Lack of Pump Control	A	III	D	30	(A) Pilot Will Detect That Pressure Gauge Is Always on High Pressure. Aircraft May Be Slightly More Responsive Also, a System Excessive Temperature Display May Occur (B) None	None - Monitor Failed System Temperature and Limit Aircraft Usage as Necessary to Control Temperature Except in an Emergency or Combat Per Developed Plan in Pilots Handbook	This Failure Will Not Pose a Hazard to Personnel or Aircraft But Could Potentially Degrade Fluid Lubricity and Overheat/Cause Pump Failure
2.0 Flow Augmentation (Actuator)	2.1 Nozzle Clogs	Unable to Achieve Maximum Actuator Rate	A	III	D	30	(A) Pilot May Detect Reduced Aircraft Performance (B) None	None - Maintain Adequate System Filtration and Maintain Acceptable Screen at Nozzle Inlet	This Failure Has Little Hazardous Effects on Personnel or Aircraft. But May Degrade Aircraft Maneuvering Performance.
	2.2 Check Valve (1) and (2) Fail Open	Same as 2.1	A	III	D	30	(A) Same as 2.1 (B) Same as 2.1	None	Same as 2.1. During Transient Operation a "Short Circuit" Between Pressure and Return Will Exist Which Limits Performance and Generates Additional Heat. However Second Check Valve Was Added to Prevent the "Short Circuit"
	2.3 Check Valve (1) and (2) Fail Closed	Same as 2.1	A	III	D	30	(A) Same as 2.1 (B) Same as 2.1	None	Same as 2.1. Assisting Load Rates Are Reduced
3.0 Load Recovery Valves (Actuator)	3.1 Check Valve (3) or (4) or Both (3) and (4) Fail Open or Closed	Same as 2.1	A	III	D	30	(A) Same as 2.1 (B) Same as 2.1	None	Same as 2.1 During Transient Operation a "Short Circuit" Between Pressure and Return Will Exist Which Limits Performance and Generates Additional Heat. For the Dwell or Small Motions Operating Time (95% of the Time) the Control Valve Will Limit Energy Loss to That Normally Experienced
4.0 Overlap Valve	4.1 No Failure Modes								
5.0 Distribution System	5.1 Fluid Leak From Fitting or Burst Tube/Hose	Will Lose Use of One Hydraulic RLS Circuit	NA	III	D	30	(A) Cockpit Indicator Light Will Turn On (B) Reservoir Level Sensing (RLS) Will Automatically Switch System Operation	None	8 000 psi CTFE Studies Indicate a Leak Will Be No More Severe Than a Leak at 3 000 psi MIL-H- 83282 Except That System Reservoir Will Deplete Faster Because of Smaller Volume Reservoir

Phase of Operation: G - Ground, T - Takeoff, F - In-Flight, L - Landing, A - All
 Hazard Classification/RHI Point Value: Class I/4 - Catastrophic, Class II/2 - Marginal, Class III/3 - Critical, Class IV/1 - Negligible
 Hazard Probability Level/RHI Point Value: A/6 - Frequent, B/3 - Reasonably Probable, C/4 - Occasional, D/3 - Remote, E/2 - Extremely Improbable, F/1 - Impossible

Figure 276
Results and Recommendations of the
Preliminary Hazard Analysis (PHA)

SECTION V
PHASE IV - COMPONENT FABRICATION AND DEMONSTRATION TESTS

Phase IV consisted of the following tasks:

- 1) Component/System Fabrication
- 2) Component/System Testing
- 3) Hydraulic Fluid Sampling

5.1 TASK 1 - COMPONENT/SYSTEM FABRICATION

5.1.1 Component Fabrication

a. Abex Pump - An Abex variable delivery, variable pressure (3,000 to 8,000 psi) pump, Model AP6VHP-3, presented in Figures 277 and 278, was delivered to MCAIR. This was the same type pump used for the FWFRHS demonstration test, except it was modified for variable pressure control. The pump had a 0.95 in³/rev. displacement, a rated delivery of 15 gpm at 4,000 rpm, 215°F inlet fluid temperature and 7,850 psi full flow pressure, suitable for use with CTFE fluid. A mounting flange per AS474 and a pilot flange per AS472B were incorporated in the design to match with the MCAIR laboratory drive motor. The ports were designed per MS33649 standard. The port sizes and locations are shown in Figure 279. The dry weight of the unit was 31 lbs.

The drive shaft was manufactured from PH17-4Mo corrosion resistant steel and provided power to the rotating group. The rotating group consisted of a cylinder barrel, a radial roller bearing, nine pistons and shoes, and a shoe retainer plate. The cylinder barrel was made of 4330 steel alloy with nine cylinder bores and the face was bronze plated. The bearings were 52100 tool steel and the pistons were M50 tool steel. The shoe faces were 4140 with bronze plate and the back end (or cup) was D2 tool steel. The shoe retainer plates were made of 4620 steel alloy and 52100 tool steel.

The shoes rode on a bearing surface called the wear plate, which was made of 52100 tool steel. The wear plate was held against the hanger made from a 4130 steel alloy casting. The pump housing was made of 356 T6 aluminum casting and to facilitate pump assembly, was designed in two halves. The port cap was made of nitralloy 135 M steel and the port plate of M50 tool steel. The stroking and rate pistons, as well as the compensator valve, were made of 52100 tool steel.

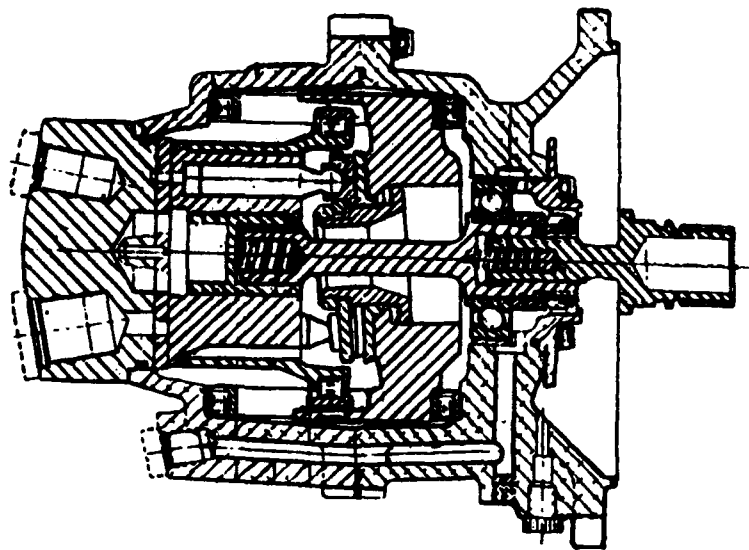


Figure 277
Variable Pressure Pump - Assembly

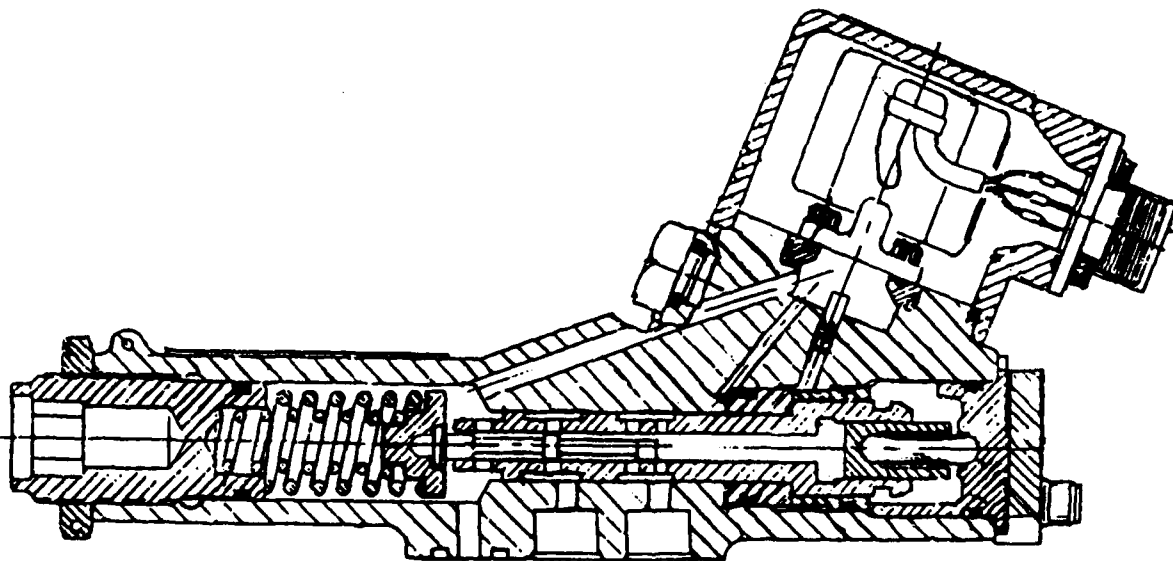


Figure 278
Variable Pressure Pump EHV

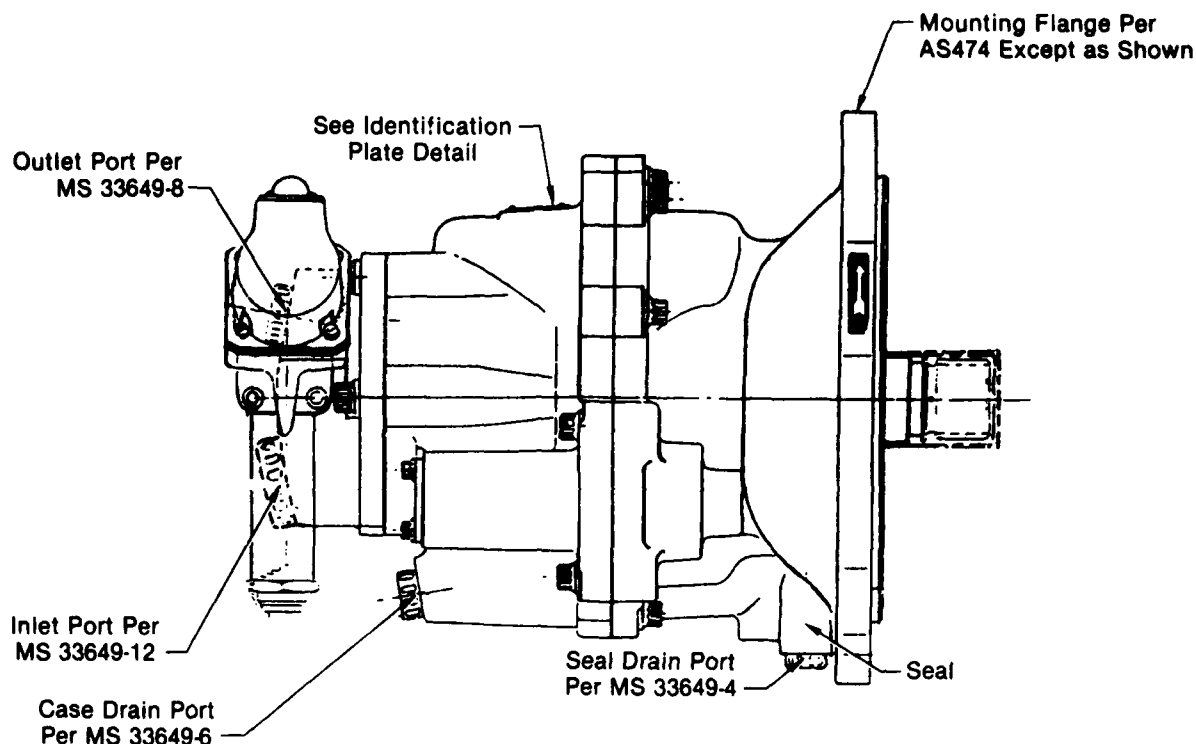


Figure 279
Variable Pressure Pump - Installation

The variable pressure pump control used a standard Abex 410 Electro-hydraulic Valve, see Figure 278, modified for special feedback spring configuration, with provisions for higher return pressures and a dropping orifice. The pump was operated open loop (no pressure feedback control), and was commanded to 8,000 psi at 0 volts and 3,000 psi at 10 volts. An electrical failure commands system pressure to 8,000 psi. These valves can be related to the 3,000 psi variable pressure pump (Abex unit) discussed in paragraph 4.1.2.

A control unit was designed and built at MCAIR to control the variable pressure. It allowed adjustment of the pressure schedule at both the 3,000 and 8,000 psi ends of the curve. The stabilator actuator main control valve position was multiplied by the servovalve flow gain to produce an estimated flow demand for the actuator. Then, a pressure schedule was used to determine the pressure setting, and a signal was sent to the pump servovalve.

b. Parker Bertea Stabilator Servoactuator - The stabilator actuator, shown in Figure 280, was selected to demonstrate the low energy consumption concepts and is a high performance, electrohydraulic servo, controlled by a two-stage, Direct Drive Valve (DDV). Two quad Linear Variable Differential Transformers (LVDT) were integrally mounted to the main ram actuator and the

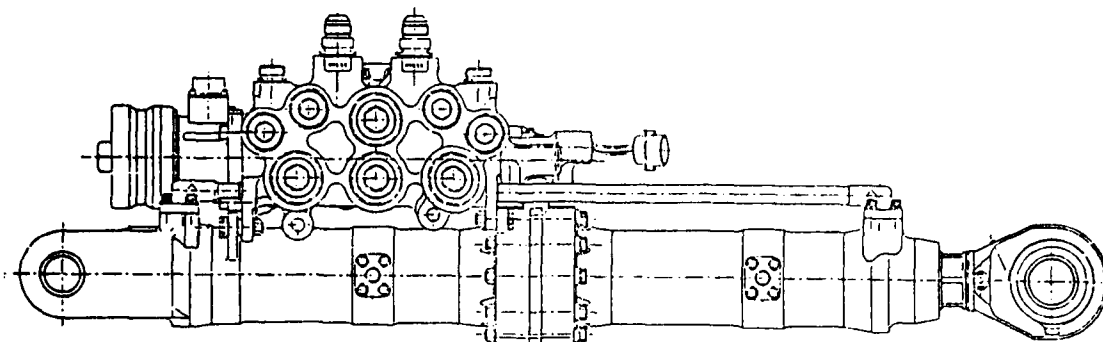


Figure 280
8,000 psi LECHT Stabilator Servoactuator

main control valve to provide loop closure. Space was provided for two flow augmentation devices and four load recovery valves in the flight weight, flight worthy Ti-6Al-4V titanium alloy valve manifold.

(1) Main Ram Assembly - The main ram (MR) assembly was a dual tandem unit with the aft system fully balanced and the forward system partially balanced. The dimensions and the areas of the actuator, shown in Figure 281, included an actuator midstroke length of 32.145 inches nominal, and a stroke of 7.770 inches nominal. The output forces at 8,000 psi inlet and 50 psi return pressures, were 43,680 pounds nominal extend and 36,875 pounds nominal retract.

	Forward Cylinder	Aft Cylinder
Cylinder Bore (in.)	2.369	2.369
Piston Rod Diameter (in.)	1.622	1.622
Guide Tube Diameter (in.)	1.242	—
Piston Extend Area (in. ²)	3.196	2.341
Piston Retract Area (in. ²)	2.341	2.341

Figure 281
Stabilator Actuator Pertinent Dimensions and Areas

The actuator cylinder and the piston assemblies were machined from 455 stainless steel, except for the center dam, which was machined from Ti-6Al-4V titanium alloy. Both piston rod glands were made of 7075-T73 aluminum alloy with hard coat anodized and honed bores; the piston rods were chrome plated and precision ground.

(2) Two-Stage Control Valve - The DDV, shown in Figure 282, consists of the force motor and the pilot valve, and is the first stage of the two-stage control valve configuration. The pilot control valve sleeve and slide was

machined from 440C stainless steel. It was driven directly by a four coil, linear electromechanical force motor, shown schematically in Figure 282, which consisted of four major components:

- o Armature assembly
- o Two pole pieces
- o Magnet assembly
- o Coil assembly

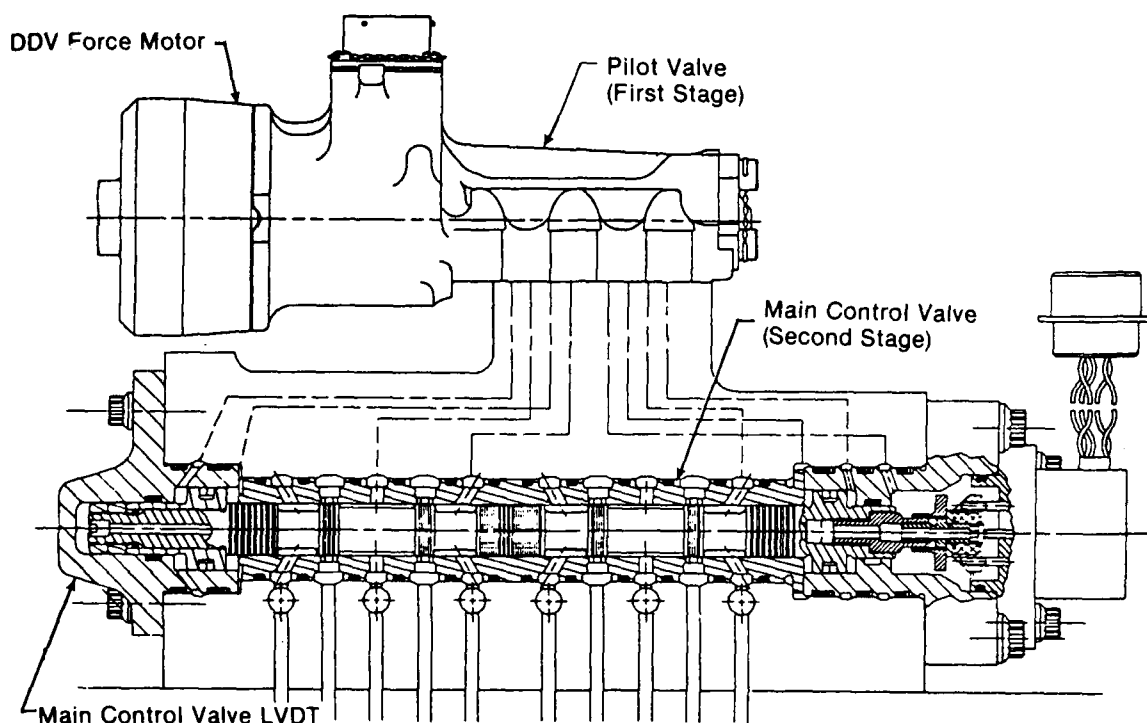


Figure 282
Direct Drive Valve and Main Control Valve Schematic

These components were combined to produce an efficient, low power, proportional drive motor with extremely low hysteresis and threshold characteristics. The motor's electrical characteristics are shown in Figure 283.

The second stage of the control valve, or the Main Control Valve (MCV) was a four-way, dual tandem assembly positioned indirectly by the first stage pilot valve. The pilot valve, connected directly to the force motor, was mounted above the MCV and was designed to control pressure and flow to the two MCV modulating piston heads. One piston head was integral with the valve spool, while the other head was attached to the main slide through a threaded joint and positively locked.

Parameter	Value
Coil Resistance Per Element	9.32 ohms
Coil Resistance Per Channel	18.64 ohms \pm 0.76
Rated Current Per Channel Quad Channel Operation*	0.25 amps
Rated Power Quad Channel Operation (0.25 amp/Channel)	1.165 watts/Channel
Rated Power Dual Channel Operation (0.5 amp/Channel)**	4.66 watts/Channel
Maximum Power for Chip Shear Operation (0.75 amp/Channel)*	42.0 watts Total
Maximum Continuous Current, Quad Channel Operation	0.5 amps/Channel

*In order to minimize power for normal operating mode, the required current per channel is 0.25 amp. For this reason, the chip shear current is three times the normal operating mode, or 0.75 amp per channel, in order to produce 48 lb at maximum travel.

**Provides full performance after two electrical channel failures

Figure 283
Force Motor Characteristics

The main stage sleeve and spool was fabricated from 440C stainless steel. The sleeve contained eight sets of metering slots, four for each system. The sleeve was sealed with O rings and backup rings in the valve manifold, which was machined from titanium.

(3) Linear Variable Differential Transducers (LVDT) - Two LVDTs were mounted integrally in the actuator to provide position feedback for the MCV and the main ram. Each LVDT, supplied by Kavlico, provided four independent channels of operation. The performance parameters of the transducer are shown in Figure 284.

The MR transducer assembly consisted of four parallel LDVTs housed in a cylindrical case manufactured from work hardened Type 304 corrosion resistant steel. The case provided the balance tube for the actuator piston when it was used internally. Two grooves on each end of the case were provided for installation of pressure seals. The case terminated at one end of the housing providing space for the electrical connector and the coil head stress relief board as well as incorporating holes for the transducer mounting.

The four transducer probes were welded to the common head after final trimming. A single dual load path, rotatable, attachment screw was swaged into the head and threaded into the LVDT arm assembly that attached to the piston at the rod end to provide a precise null adjustment.

	Main Ram	Main Control Valve
Excitation	8 VRMS \pm 0.5%, 1,833 Hz \pm 5%	8 VRMS \pm 0.5% 1,833 Hz \pm 5%
Maximum Power Input	0.041 to 0.058 VRMS Each Channel at 8 VRMS and 68°F Power Factor 0.74 to 0.85	0.19 to 0.23 VRMS Each Channel at 8 VRMS and 68°F Power Factor 0.3 to 0.5
Normal Output Voltage	\pm 5 VRMS	\pm 3 VRMS
Accuracy	\pm 0.075 VRMS Maximum Deviation From the Straight Line Defined by the Scale Factor at 68°F Within \pm 3.5 in. From the Null Position. \pm 0.085 VRMS For the Remaining Parts of the Stroke.	\pm 0.035 VRMS Maximum Deviation From the Nominal Gain Throughout the Electrical Stroke of \pm 0.035 in. at 70°F.
Tracking	0.060 VRMS Maximum Difference Between Any Two LVDTs Within \pm 3.5 in. Over the -40 to 275°F Temperature Range. May Not Exceed 0.070 VRMS for the Remaining Parts of the Stroke.	0.065 VRMS Maximum Difference Between Any Two LVDTs Over the Operating Temperature Range and Operating Stroke of \pm 0.035 in.
Null Voltage	0.025 VRMS Maximum Over the -40 to 275°F Temperature Range	0.035 VRMS Maximum Over the -40 to 275°F Temperature Range
Scale Factor Temperature Coefficient	-1.0% Per 100°F Rise	-2.0% Per 100°F Rise
Load Impedance	40,000 to 44,000 ohms	40,000 to 44,000 ohms
Scale Factor	1.246 (in Phase) VRMS/in. at 68°F	42.36 VRMS/in. at 68°F
Phase Shift	10° \pm 3°	3.5° \pm 3.5°
Phasing	With Pins 2 and 3, 7 and 8, 12 and 13, 17 and 18 Interconnected Commonly and Excitation Signal Applied to Pins 1, 6, 11 and 16. The Voltage on All Other Pins Shall Be in Phase With the Excitation When the Probe Is Displaced From Neutral Toward Its Attachment Point.	With Pins 2 and 3, 7 and 8, 12 and 13, 17 and 18 Interconnected Commonly and Excitation Signal Applied to Pins 1, 6, 11 and 16. The Voltage on All Other Pins Shall Be in Phase With the Excitation When the Probe Is Displaced From Neutral Away From Its Attachment Point.

Figure 284
Transducer Performance Parameters

The main control valve LVDT was similar in design to the main ram unit, except it had a smaller stroke (\pm 0.040 vs. \pm 4.10) and was overall, a smaller unit. Both LVDTs were designed to an operating pressure of 1,500 psi, with proof and burst pressures of 3,000 and 4,500 psi respectively.

5.1.2 System Description and Instrumentation - The hydraulic system used to demonstrate the low energy consumption concepts is shown schematically in Figure 260. An Abex 8,000 psi, 15 gpm intelligent pump, supplied pressure and flow to the system. An electrohydraulic valve (EHV) was used to drive the pump compensator position to a fixed outlet pressure establishing baseline performance values. In addition, the EHV controlled the compensator when operating in the variable pressure mode.

An 8,000 psi version of the F-15 stabilator servoactuator, supplied by Parker Berteau, tested the overlap valve, flow augmentation and load recovery valve concepts. The interconnecting lines, filters and reservoir represented

an aircraft installation. Aircraft Porous Media (APM) provided a three-micron pressure filter and a five-micron return filter. The reservoir was a Metal Bellows Corporation (MBC) 8,000 psi bootstrap type, with a 4,000 psi accumulator providing constant pressure. Appropriate check valves, relief valves and a heat exchanger were also included.

The test setup was enclosed in a special thermal control chamber, which was used for the Flight Worthiness of Fire Resistant Hydraulic Systems (FWFRHS) demonstration system test. The chamber was capable of providing ambient temperatures from -65°F to 160°F. CTFE fluid temperatures were regulated by adjusting water flow through the hydraulic fluid to a heat exchanger. An Enerpac 10,000 psi pump with low flow capability was used to fill and bleed the hydraulic system open loop.

The stabilator servoactuator was mounted into a test fixture designed to simulate aircraft geometry, inertia, stiffness and load conditions. The load system for the test unit included a load actuator, an accumulator to set the load and a ground cart capable of supplying MIL-H-5606 hydraulic fluid at 3,000 psi pressure to activate the load system.

The LECHT test system was instrumented as shown in Figure 260. A Soltec transient recorder Model SMR with an accuracy of ± 0.025 percent full scale was utilized. The instrumentation transducers had accuracies as follows:

- o Pressure $\pm 1/2$ percent full scale
- o Temperature $\pm 2^\circ\text{F}$
- o Load cells ± 3 percent full scale
- o LVDT $\pm 1/2$ percent full scale
- o Flow ± 1 percent full scale (turbine flowmeter)
- o Pressure gages ± 1 percent full scale

An Acurex Autodata 1200A torsion measurement system measured pump torque with the following accuracies:

- o Torque ± 1 percent full scale
- o RPM ± 0.25 percent full scale
- o HP ± 1.5 percent full scale

The error signal conditioners for all the above parameters were 2 percent, except for the flow conditioners which were only 0.2 percent.

Data was recorded using a Neff differential multiplexer digital data recording system with accuracies of ± 0.05 percent and ± 0.003 percent/ $^\circ\text{C}$. Data was plotted on a Versatec V-80 printer/plotter. An eight channel recorder with an accuracy of 1 percent full scale was available. A Bafco Model 916 recorder was utilized for frequency response testing. The amplitude accuracy was ± 0.1 dB and phase accuracy is ± 0.75 degrees. Data was plotted on a Hewlett Packard Model 7047 plotter with an accuracy of ± 0.5 percent.

5.2 TASK 2 - COMPONENT/SYSTEM TESTING

5.2.1 Component Verification and Testing

a. Overlapped Valve - The overlapped valve testing was evaluated for its effect on valve leakage, servoactuator frequency response, threshold and system heat exchanger size.

The primary benefit of overlapped MCV spools is the reduction in null leakage which ultimately translates into reduced steady-state horsepower. Figure 285 shows a comparison of null leakage at 160°F and 275°F fluid temperature. Two different 0.003-inch overlapped spools were

Valve Configuration	Leakage Rate (gpm)			
	Fluid Temperature (°F)	System Pressure		
		8,000	5,500	3,000
Line-to-Line	160	0.84	0.70	0.50
	275	0.85	0.77	0.52
0.003 in. Overlap No. 1	160	0.18	0.20	0.18
	275	N/A	N/A	N/A
0.003 in. Overlap No. 2	160	0.32	0.39	0.33
	275	0.44	0.50	0.34

Figure 285
Comparison of Actuator Null Leakages
With Various MCV Configurations

evaluated during the testing, with the first (labeled "OVERLAP #1" in Figure 286), lost due to a shipping error. The second overlapped valve ("OVERLAP #2" in Figure 286), exhibited higher steady-state null leakage than the first.

A 62 percent reduction at 8,000 psi and a 34 percent reduction at 3,000 psi was realized with the second overlapped spool vs. line-to-line. This variation was due to the fact that there was no reduction in leakage at 3,000 vs. 8,000 psi with the overlapped valve. This apparent anomaly was likely due to "clamping." A pressure differential across the return port caused the sleeve to clamp down against the spool, and reduced the clearance which resulted in a lower leakage. As the system pressure was reduced from 8,000 psi, the clamping effect was lessened and leakage increased slightly. Combined with the normal decrease in leakage due to lower pressure, the total leakage remained about the same, or increased slightly. The first of the 0.003-inch overlapped valves installed in the actuator, showed a significantly improved reduction in leakage. At 3,000 psi, there was a 64 percent reduction while at 8,000 psi, a 78 percent reduction occurred.

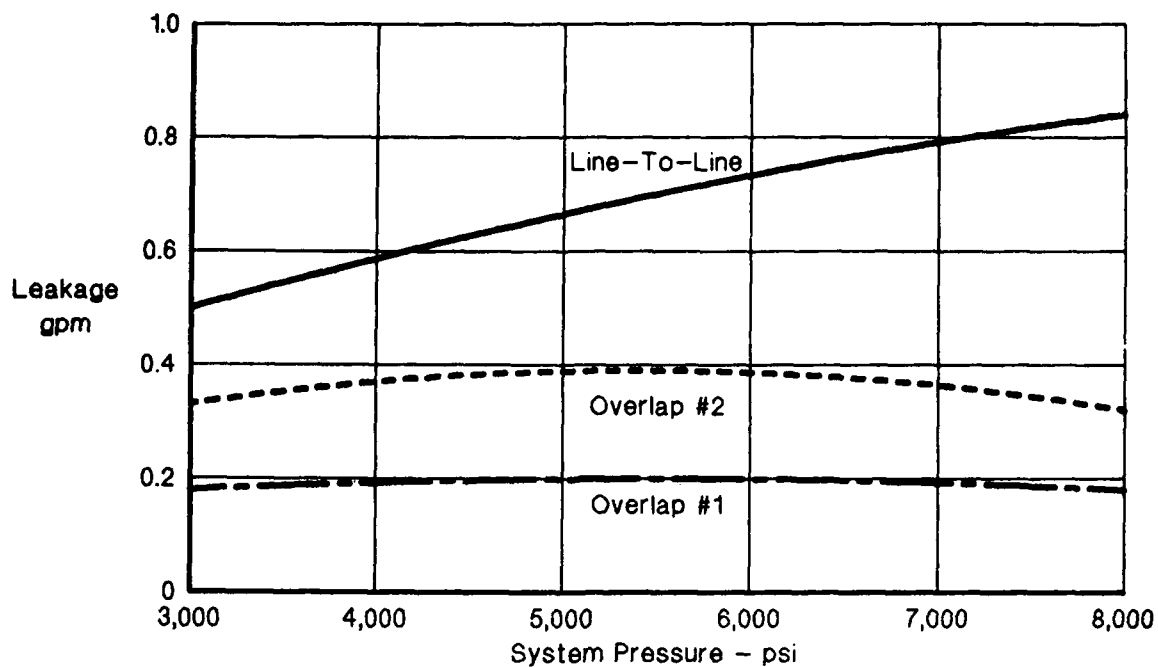


Figure 286
Plot of Null Leakage vs. Pressure for
Various MCV Configurations

Figure 287 shows a comparison of pump power required at 8,000 psi quiescent leakage conditions for PC-1 and PC-2. Regardless of what valve was used, it took 10.25 hp per 15 gpm pump at 3,800 rpm. Combined with the affects of the actuator leakages, assuming ten flight control actuators per aircraft (eight tandem and two simple), a total of 55.6 hp was required for line-to-line systems and 28.0 hp was required for overlapped valve systems. This was nearly a 50 percent reduction in power consumption with the reduced null leakage of 0.003-inch overlapped spools. If the higher leakages of the second overlapped valve leakages were used, 33.8 hp would be required (39 percent reduction).

In a variable pressure system, system pressure during periods of low actuator movement (when null leakage is most important), would be 3,000 psi. Using 5.9 hp per 15 gpm pump at 3,000 psi and 3,800 rpm, a total of 19.5 hp for line-to-line systems is required. With the first overlapped valve, system power would drop to 14.6 hp (25 percent reduction), and with the second overlapped valve, power dropped to 16.9 hp (13 percent reduction).

From an actuator performance standpoint, there is a penalty with overlapped valves. The deadband tends to make the valve unresponsive to small signals. This can be compensated for by designing a high gain servo-valve. When comparing the 0.003-inch overlapped valve with the line-to-line valve, the only performance difference noted was in small amplitude frequency response tests (± 1 percent full stroke), and threshold tests. Although the line-to-line valve was an order of magnitude better in threshold testing, both valves were within F-15 stabilator specifications.

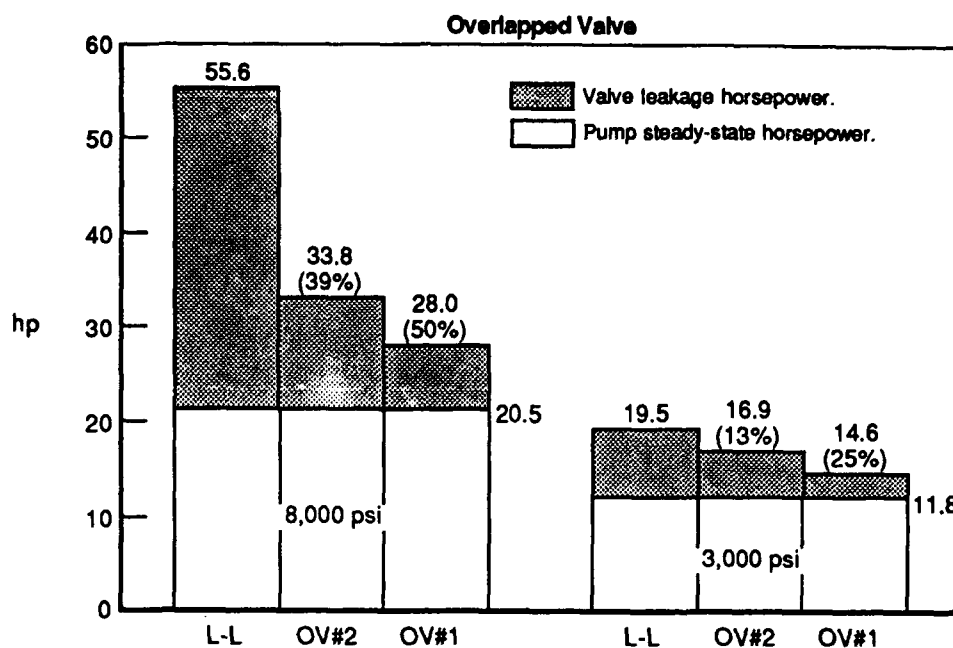


Figure 287
Steady-State Pump Horsepower for
Various MCV Configurations

Figures 288 through 290 show plots of no-load 1, 2 and 10 percent amplitude frequency responses at 8,000 psi with the overlapped valve installed as compared to the line-to-line valve installed. For the overlapped valve, the 1 percent amplitude was down 3 dB at 0.8 Hz, and was outside specification values. At 2 percent, the problem had diminished, and for the most part, the actuator was within specification limits. The 10 percent amplitude was well within specification limits. As seen in Figure 288, the line-to-line valve performed better at low amplitudes. However, at higher amplitudes, as seen in Figure 290, the overlapped valve performed better.

b. Flow Augmentation and Load Recovery Valves - The comparison of conventional system performance with flow augmented system performance is presented in Figures 291 and 292. Figure 291 shows the no-load performance comparison. The rates were comparable, 8.3 vs. 8.5 in./sec for extend and 8.5 in./sec for both in the retract direction. The data showed an extend flow reduction of 56 percent and retract flow reduction of 32 percent. The average of 44 percent reduction was reasonably close to the design goal of 50 percent reduction.

Figure 292 presents a performance comparison when applying an assisting load equal to 89 percent of the actuator's stall load capability. Here, the extend direction showed a 63 percent reduction in central system flow demand using the Flow Augmented/Load Recovery Valve (FA/LRV) system. In the retract direction, the reduction was 49 percent using the FA/LRV system. The combined two direction benefit was 56 percent. It should be noted that increase in central system flow demand with the assisting load over the no-load condition was 52 percent for the conventional system and only 19

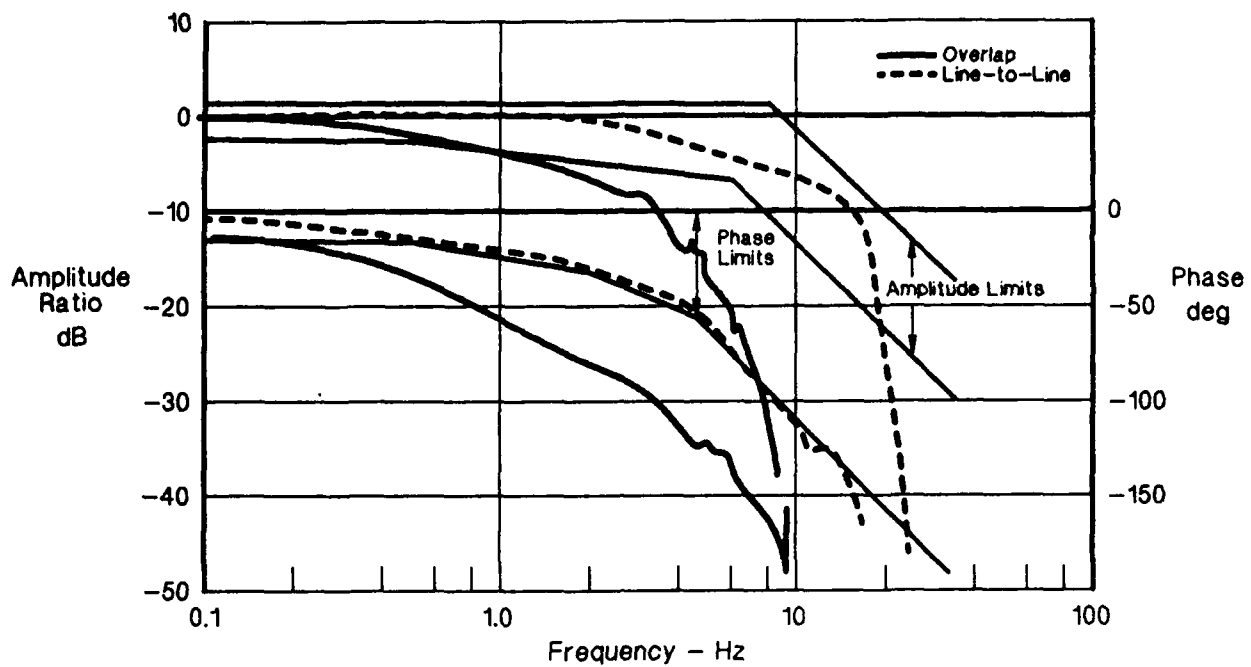


Figure 288
No-Load, 1% Main Ram Frequency Response
Overlapped vs. Line-to-Line - 8,000 psi

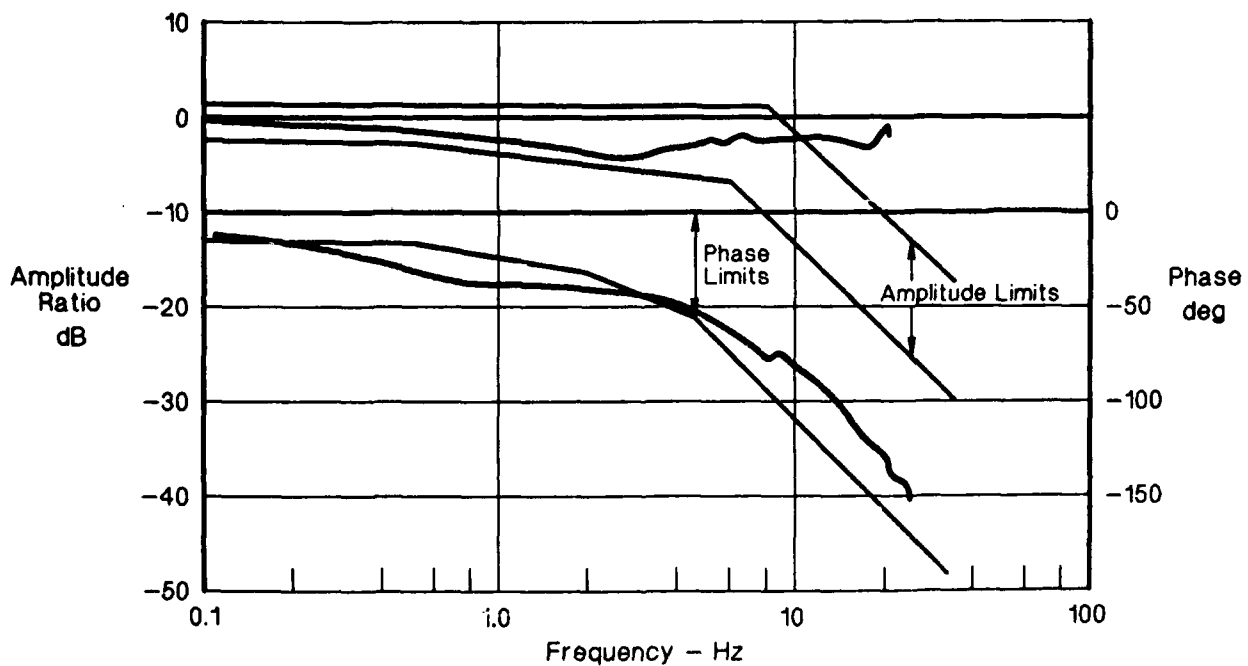


Figure 289
No-Load, 2% Main Ram Frequency Response
Overlapped Valve - 8,000 psi

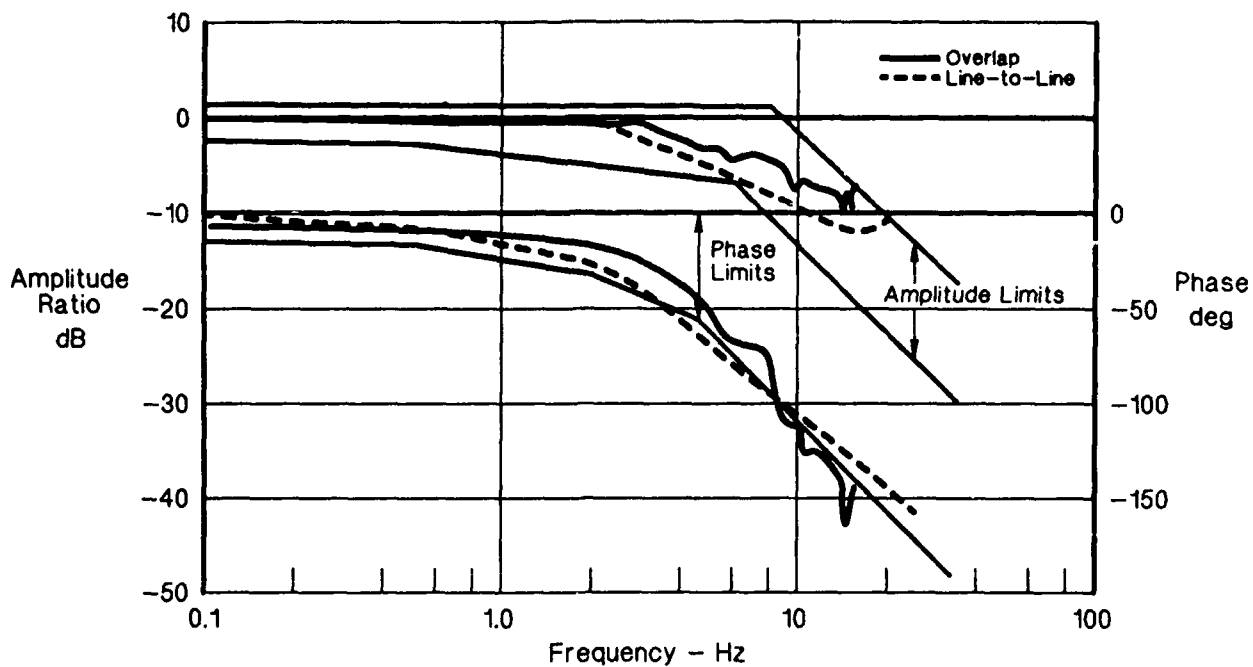


Figure 290
No-Load, 10% Main Ram Frequency Response
Overlapped vs. Line-to-Line - 8,000 psi

Condition	Conventional System		Flow Augmentation System		Percent Flow Reduction
	Flow (gpm)	Rate (in./sec)	Flow (gpm)	Rate (in./sec)	
Extend Direction	9.1	8.5	4.0	8.3	56
Retract Direction	9.5	8.5	6.5	8.5	44
Average Reduction (Both Directions)					44

Figure 291
No-Load Comparison
Conventional System vs Flow Augmentation

Condition	Conventional System		Flow Augmentation System		Percent Flow Reduction
	Flow (gpm)	Rate (in./sec)	Flow (gpm)	Rate (in./sec)	
Extend Direction	13.5	10.5	5.0	30.0	63
Retract Direction*	14.7	11.0	7.5	12.5*	49
	Average Reduction (Both Directions)				56

Figure 292
Fully Loaded Comparison
Conventional System vs Flow Augmentation

percent for the FA/LRV system. This verified the analytical predictions which indicated that the FA/LRV concepts would significantly reduce central system assisting load flow demands.

Load Recovery Valve Results - Figures 293 and 294 present analytical predictions and test results for the actuator extend and retract direction of operation. Both FA/LRV and conventional system performance are presented.

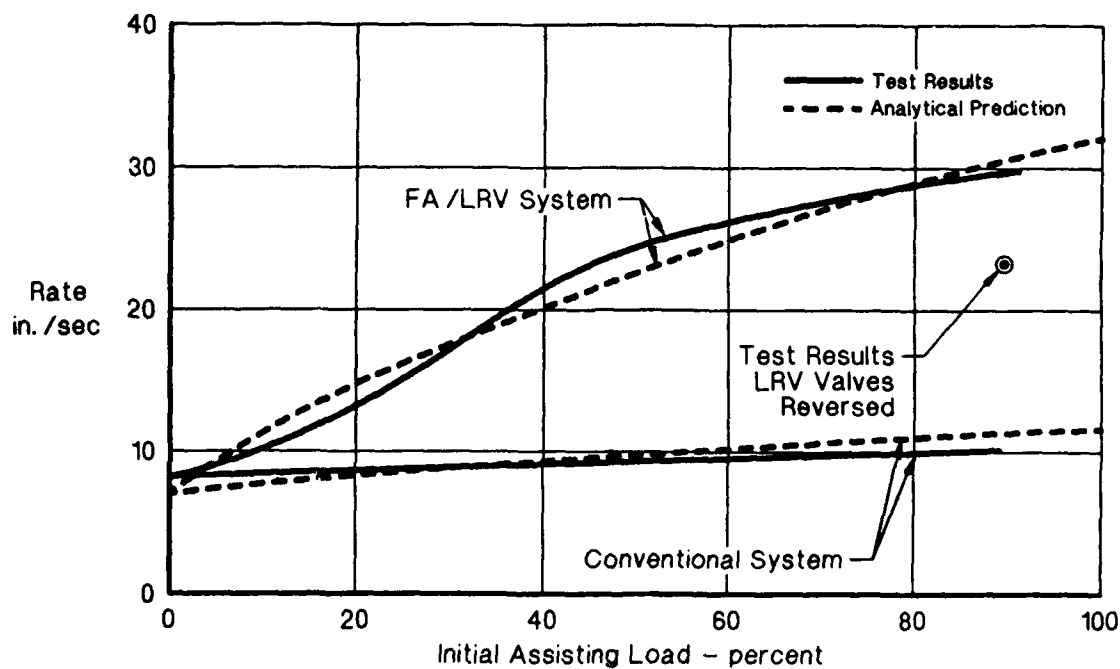


Figure 293
Actuator Extend Direction, Assisting
Load Performance Comparison

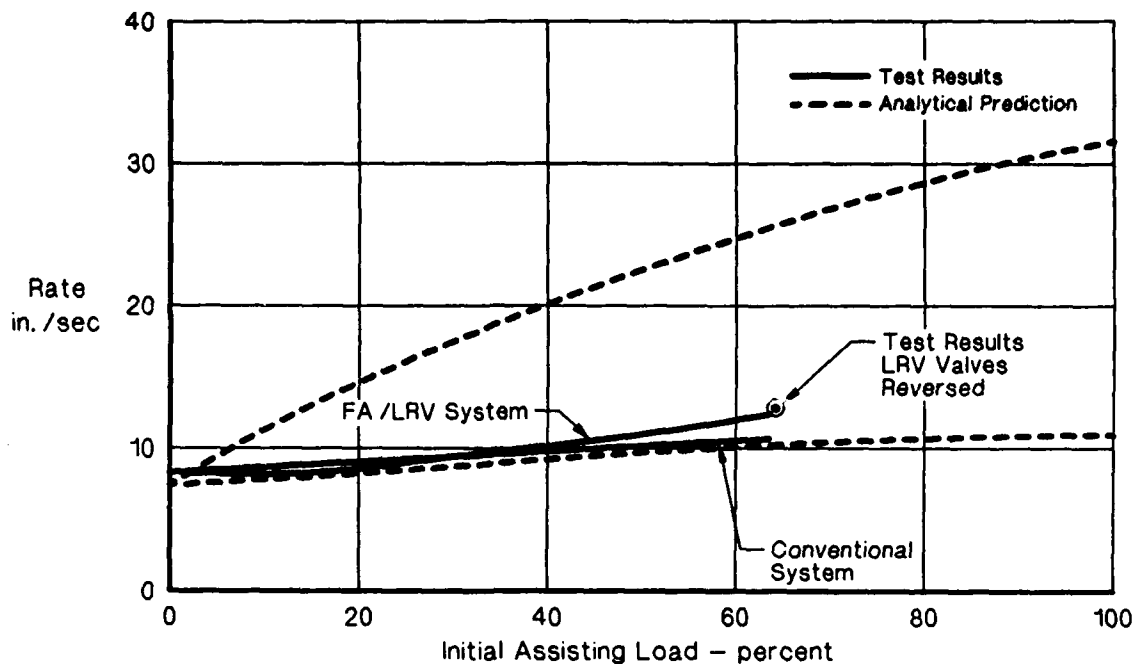


Figure 294
Actuator Retract Direction, Assisting
Load Performance Comparison

In the extend direction, the analytical prediction vs. test data showed reasonable correlation. At 89 percent assisting load, the FA/LRV system rate was 286 percent greater than the conventional system rate. This correlates very closely to the predicted differential.

The test data for the actuator retract direction of operation indicated that the load recovery valve function was not operating. The predicted rate increase at 62 percent assisting loads was approximately 2.46 times. The measured increase was only 14 percent. The reasons for the malfunction were believed to be either an LRV sticking closed, an internal passage not correctly machined, or restricted flow.

The valve function was checked by reversing the positions of the LRVs (see Figure 295). In other words, the two valves associated with the extend direction of operations, were removed and replaced with valves associated with the retract direction of operation. The tested good extend valves were then installed in the retract direction function. The 89 percent loaded extend and 62 percent loaded retract test conditions were executed. The results are shown in Figures 293 and 294. The extend direction LRV function showed a reasonable rate increase, 22 in./sec vs. 30 in./sec for the original testing. The improper functioning of the FA/LRV in the retract direction remained. The 13.0 in./sec rate was almost identical to the original 12.5 in./sec measured performance. Therefore, the test results eliminated sticking valves as the problem.

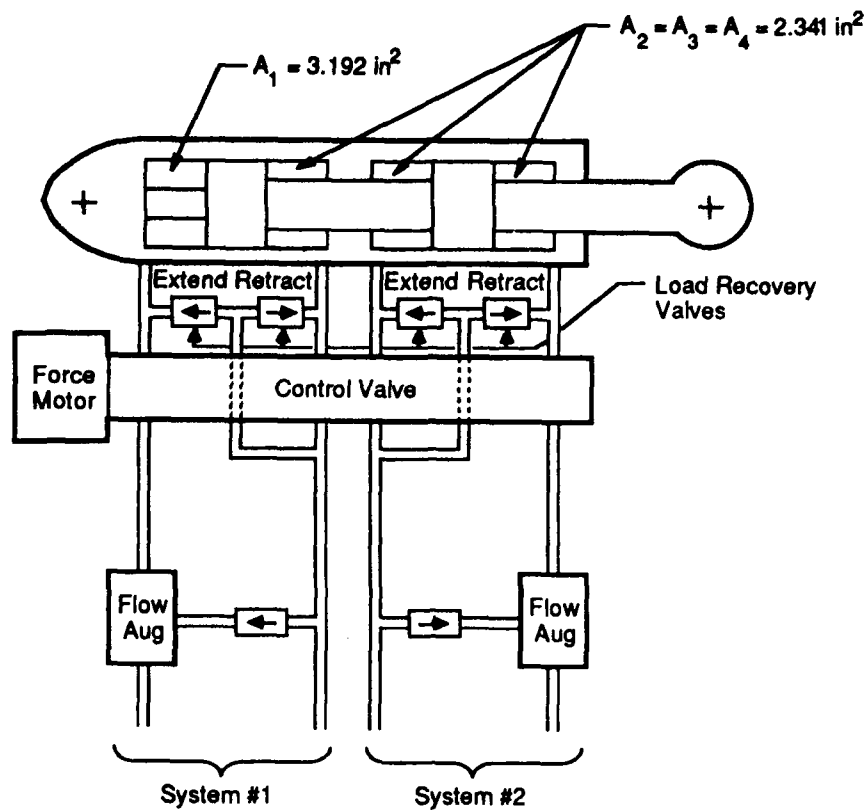


Figure 295
8,000 psi LECHT Stabilator Actuator Schematic

Flow Augmented Servo Valve Technology (FAST) Actuator Results combined FA/LRV, Figures 296 through 300 present the combined concepts performance comparison. A step command was input in each direction to establish the actuator/system rate capability. Figure 296 shows the no-load performance comparison. Both systems had much the same performance (rate capability and time to position), as shown. Even at no-load, the load recovery valves provided a higher rate transiently as can be seen in the actuator extend performance, 10.5 in./sec. vs. approximately 8.0 in./sec. Figure 297 presents the 89 percent peak load performance comparison. In the extend direction with the load recovery concept functioning, the FA/LRV concept performance was superior. The peak rate was much higher and the commanded position was achieved in approximately 90 percent of the time required for a conventional system. The FAST actuator system achieved the no-load position in approximately 42 percent of the time required by the conventional system. In the retract direction (see Figure 298), the conventional system time to commanded position was faster because the LRV concept apparently was not working and the average rate of the flow augmentation was, by design, slower. Figures 299 and 300 present the FA/LRV concepts performance vs. various peak assisting loads. Figure 299 presents the extend direction performance. As the assisting load reduced, the performance envelope approached the no-load envelope. Figure 300 shows the retract direction performance and orderly transition from maximum load to zero load.

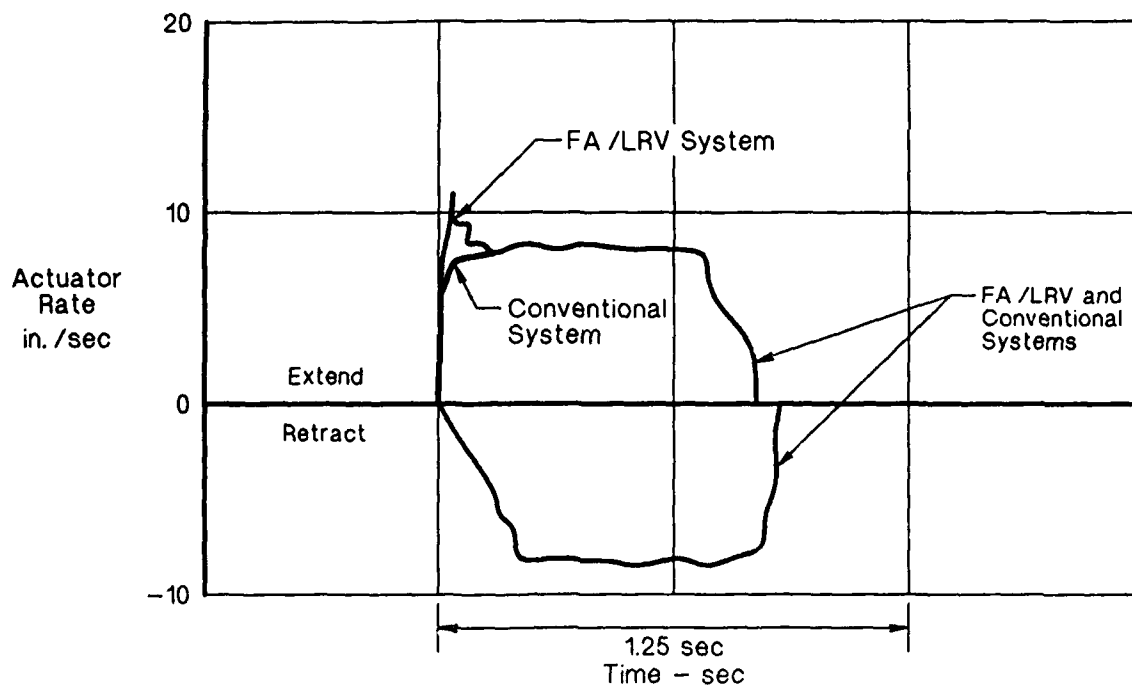


Figure 296
No-Load Performance Comparison,
FA/LRV System vs. Conventional System

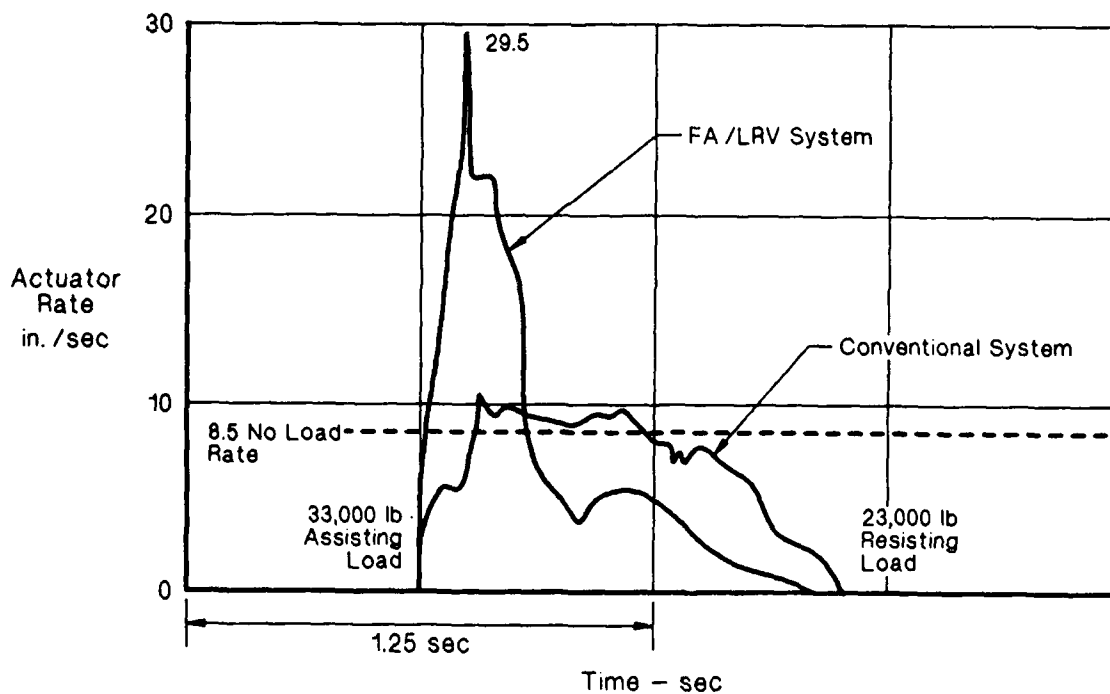


Figure 297
89% Load Extend Performance Comparison,
FA/LRV System vs. Conventional System

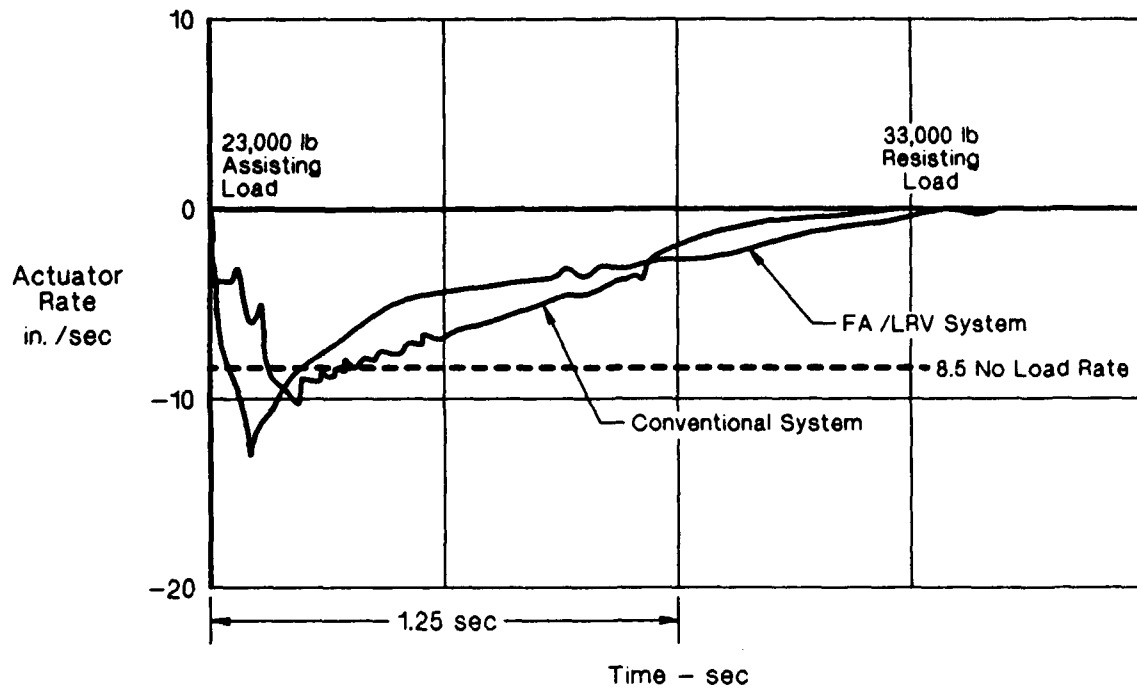


Figure 298
62% Load Retract Performance Comparison,
FA/LRV System vs. Conventional System

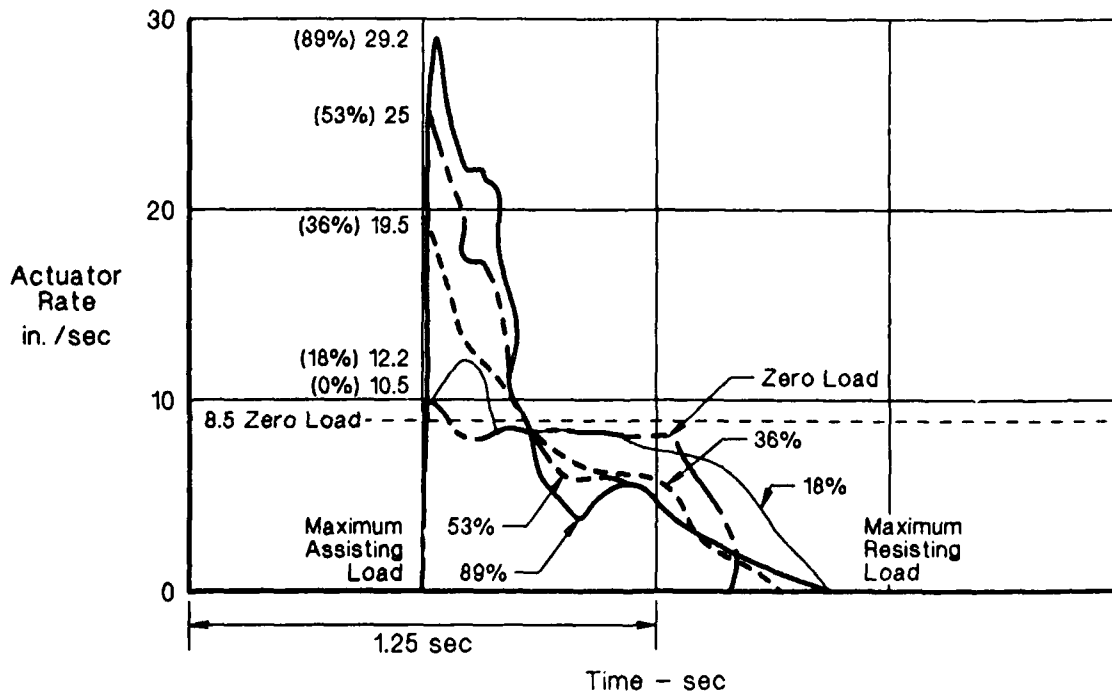


Figure 299
FA/LRV System Extend Performance Comparison
for Various Assisting Loads

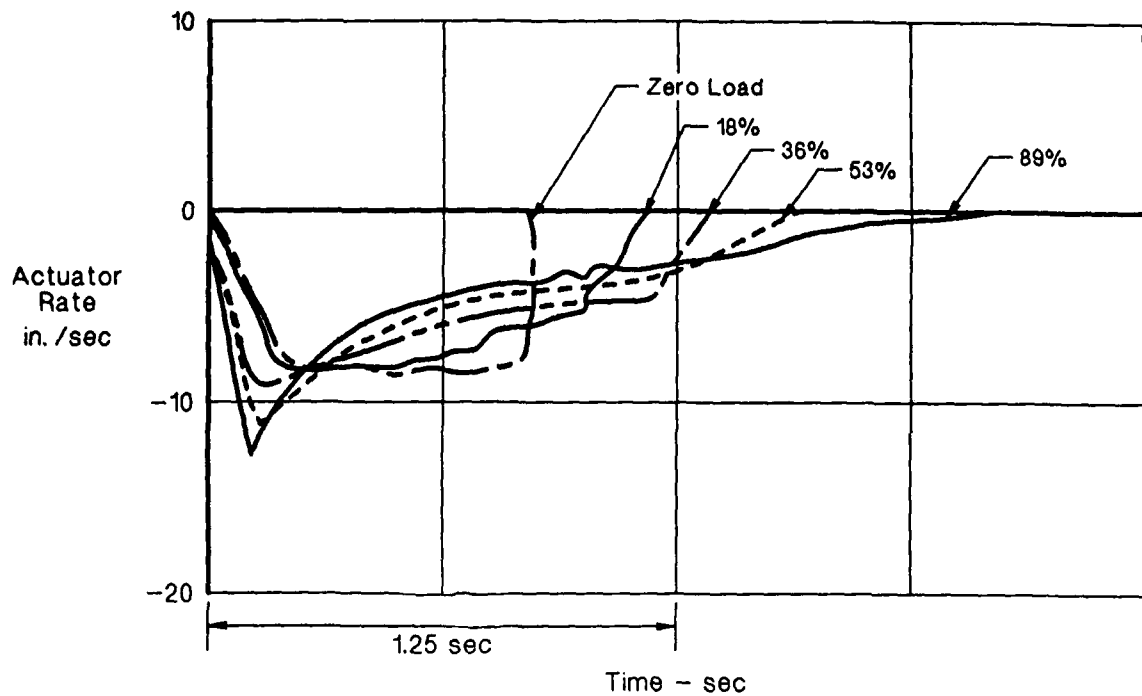


Figure 300
FA/LRV System Retract Performance Comparison
for Various Assisting Loads

Detailed Comparison with Conventional System - Figures 301 through 310 present test results for no-load operation of the FA/LRV system. Retract-extend-retract operation is shown. Figure 301 presents data from a pressure transducer downstream of the pump. The pressure varied from a peak of 8,330 psi to a minimum of 7,600 psi as the actuator stopped and started. Figures 302 and 303 present pressures immediately upstream and downstream of system 1, the unbalanced system (see Figure 295). A 500 to 600 psi drop in the lines occurred, with well controlled transient peaks, below 8,500 psi. Figure 303 shows return pressures and the variation in steady-state drop associated with the unbalanced piston areas in system 1. Also, the transient from releasing the compressed energy in the cylinder on valve opening should be noted; 2,500 to 3,000 psi added to the steady-state return flow resistance.

As discussed earlier, most of the no-load losses occurred in the actuator, jet pumps, valve and manifold passages, as required for optimum flow augmentation. Figures 304 and 305 present the balanced system 2 pressures immediately upstream and downstream of the actuator. The pressure and return side steady-state loss was the same in both directions, since the piston areas were equal. Return valve opening transients of approximately 2,000 psi above the steady-state pressures were noted.

The pressure side drop in system 2 was much higher than system 1, 3,000 psi vs. 500 to 600 psi. Identical jet pumps were used in both systems. Apparently, the two jet pumps were too large for the system 2 side. Additional resistance was required in the pressure side to limit no-load rates to the desired 8.5 in./sec. Figure 306 shows the return flows for each direction of operation. Figure 307 presents the valve position in volts and Figure 308 presents the MR position in inches. The total stroke was 5.8

inches, well under the maximum of 7.7 inches. Figure 309 shows the actuator rate in inches per second for the retract and extend directions of operation. Figure 310 presents the energy consumption in in.-lbf, which is pump torque integrated over time.

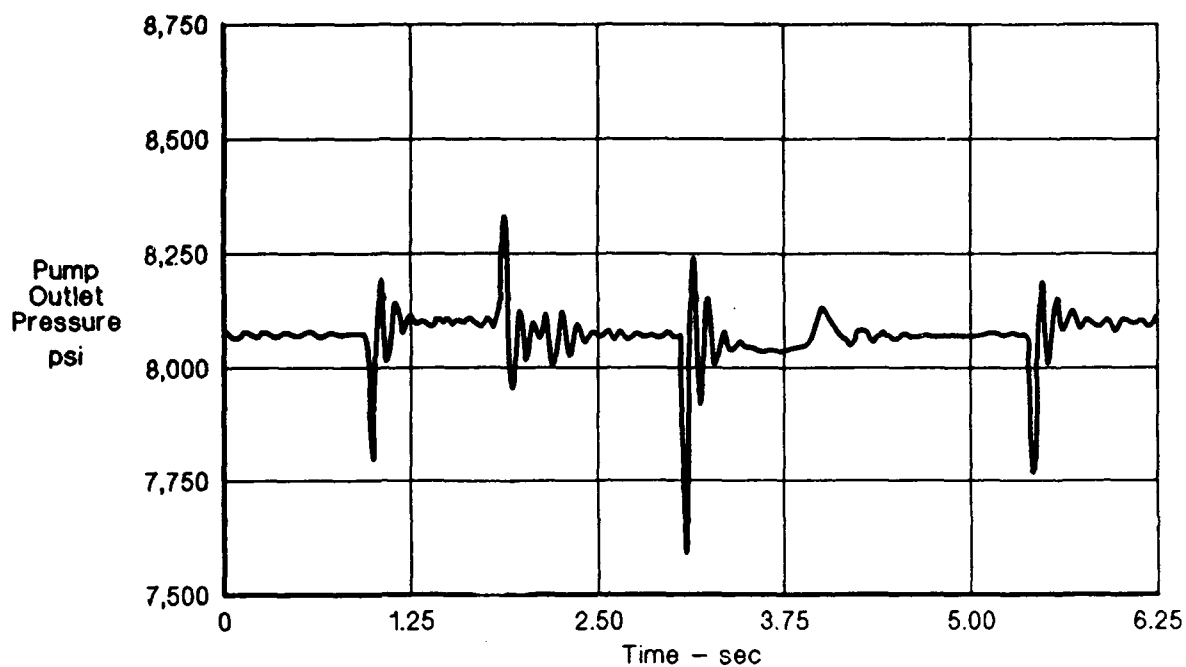


Figure 301
No-Load Step Response - 8,000 psi - Full Up
Configuration - Pump Outlet Pressure

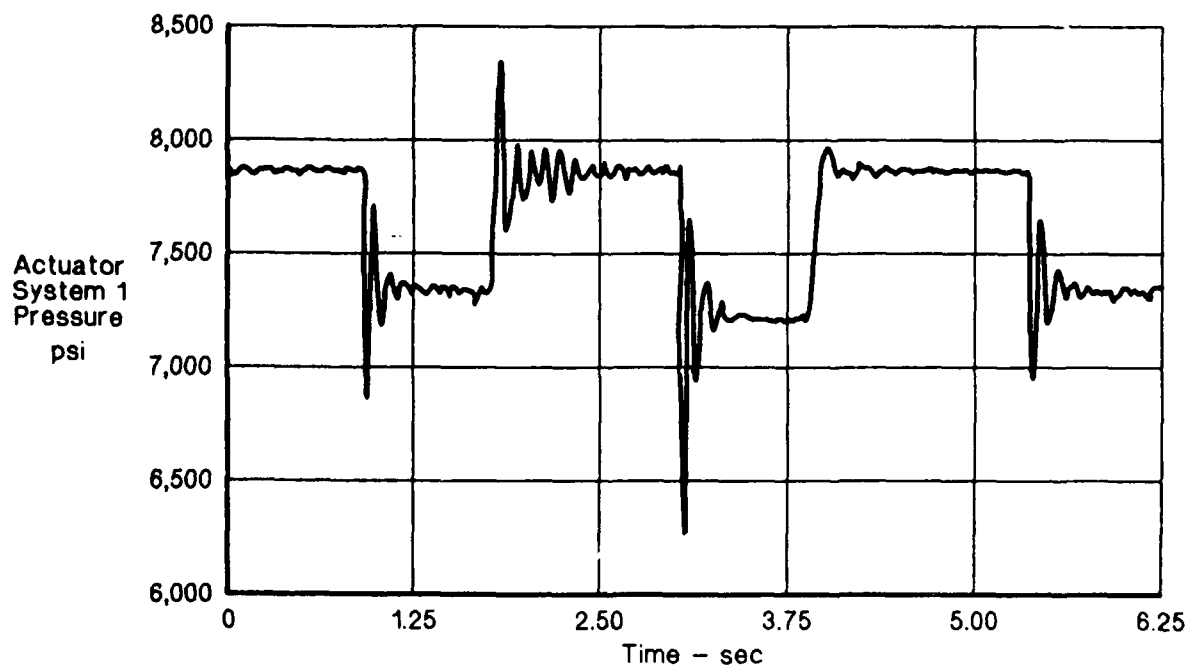


Figure 302
No-Load Step Response - 8,000 psi - Full Up
Configuration - System 1 Pressure

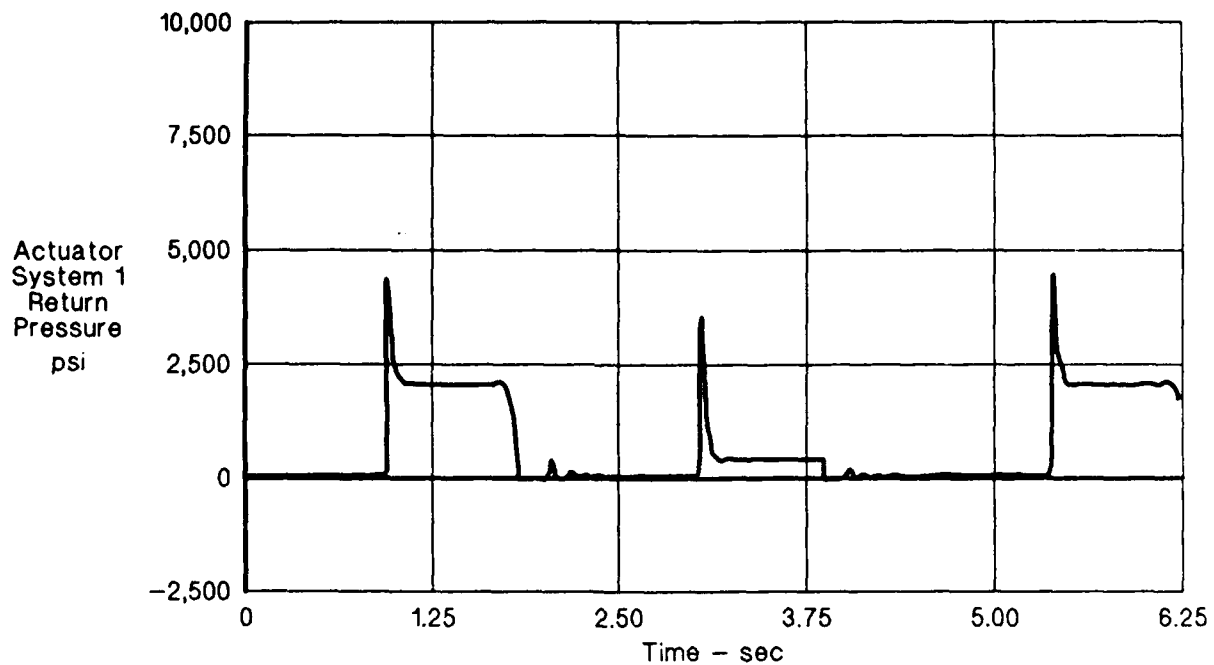


Figure 303
No-Load Step Response - 8,000 psi - Full Up
Configuration - System 1 Return Pressure

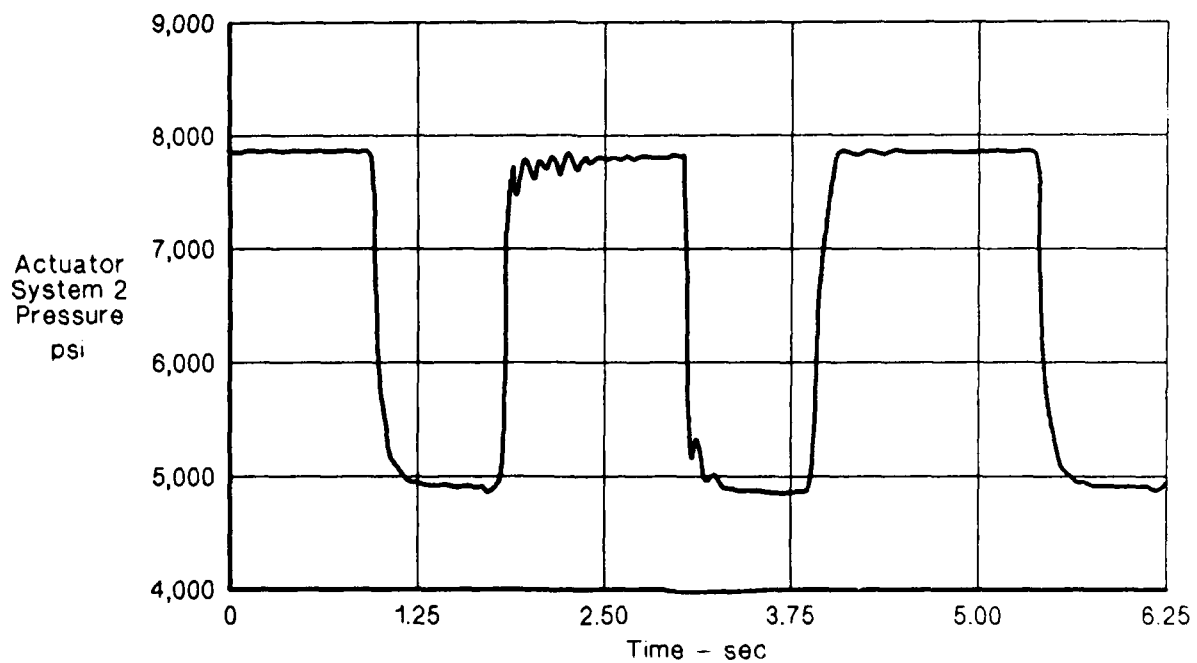


Figure 304
No-Load Step Response - 8,000 psi - Full Up
Configuration - System 2 Pressure

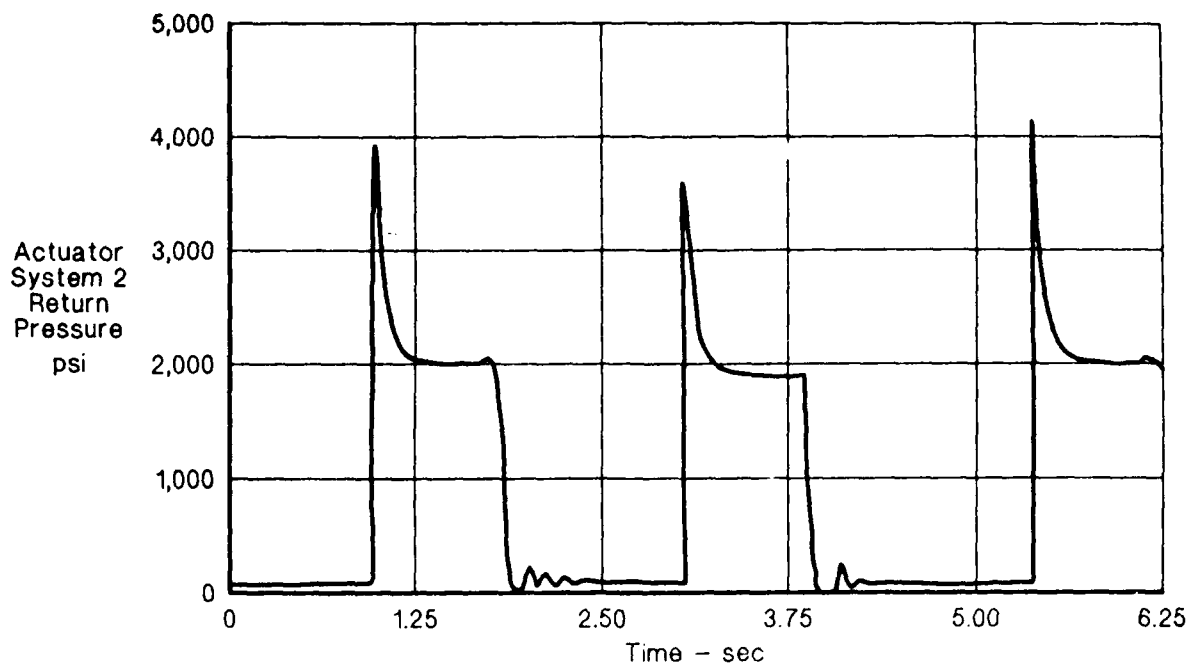


Figure 305
No-Load Step Response - 8,000 psi - Full Up
Configuration - System 2 Return Pressure

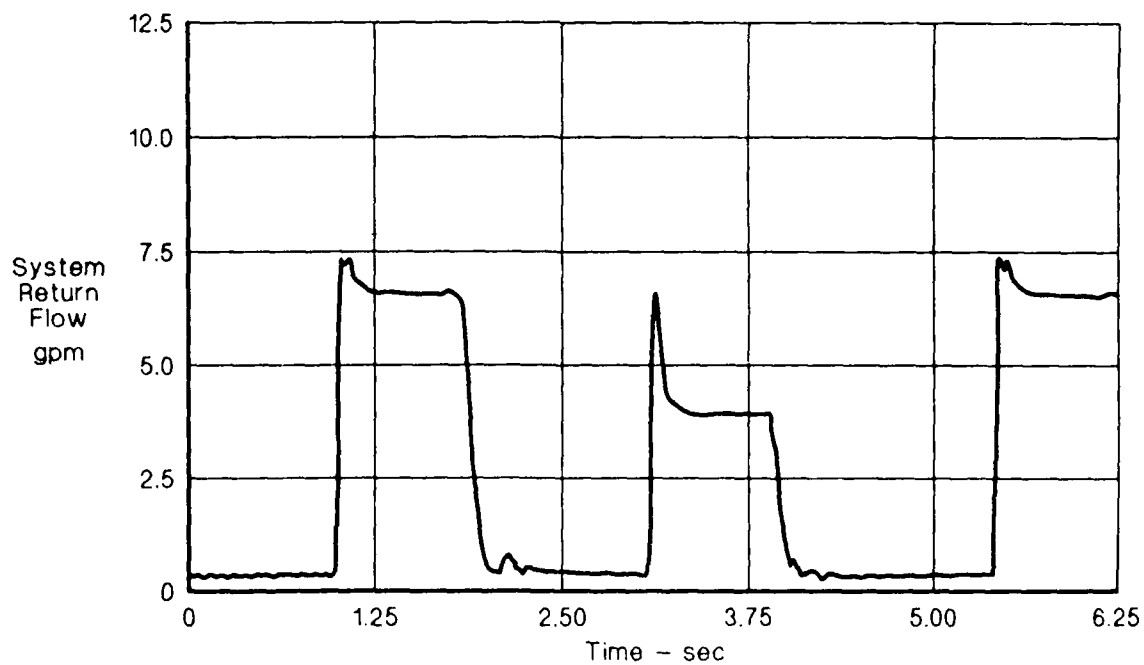


Figure 306
No-Load Step Response - 8,000 psi - Full Up
Configuration - System Return Flow

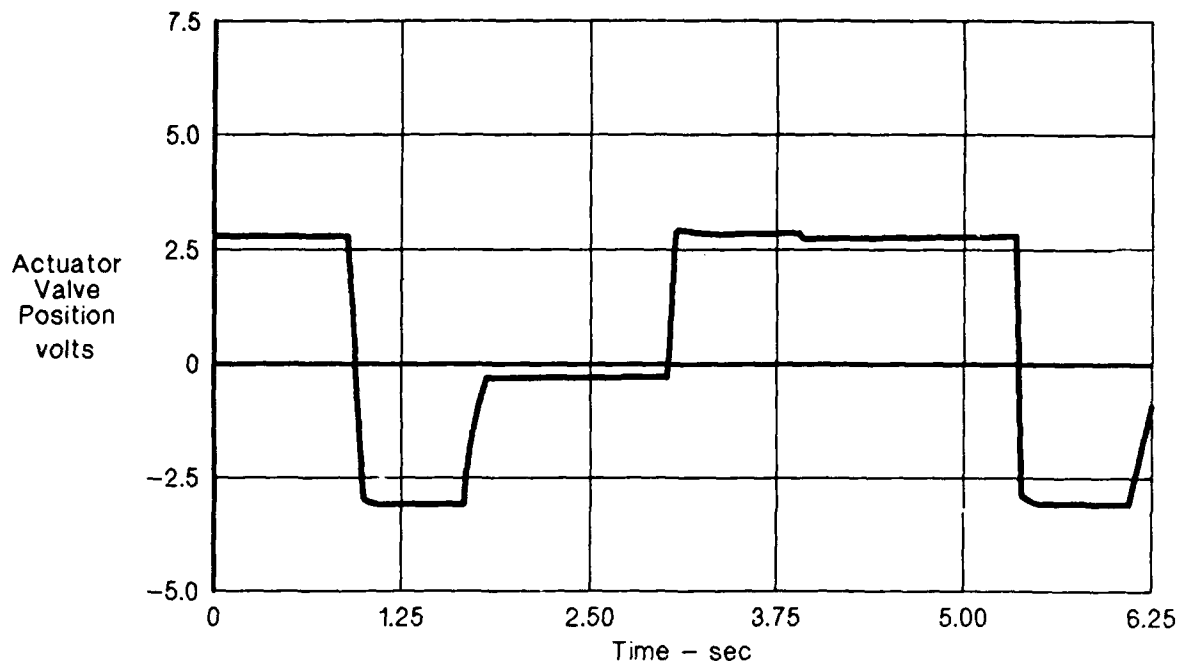


Figure 307
No-Load Step Response - 8,000 psi - Full Up
Configuration - Actuator Valve Position

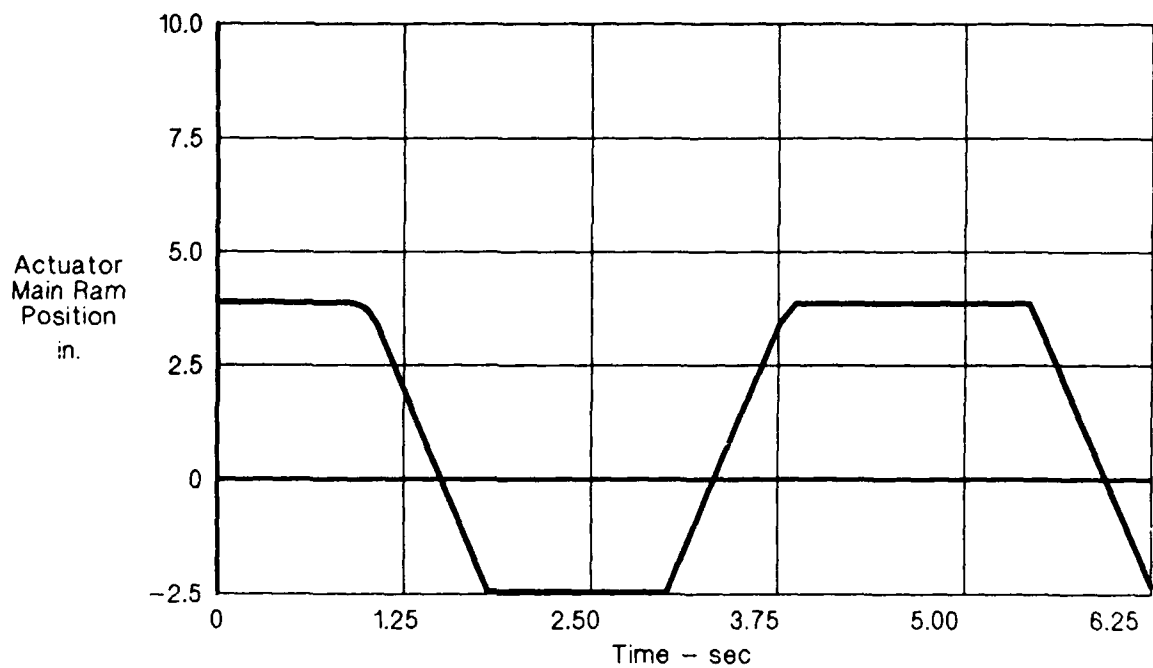


Figure 308
No-Load Step Response - 8,000 psi - Full Up
Configuration - Main Ram Position

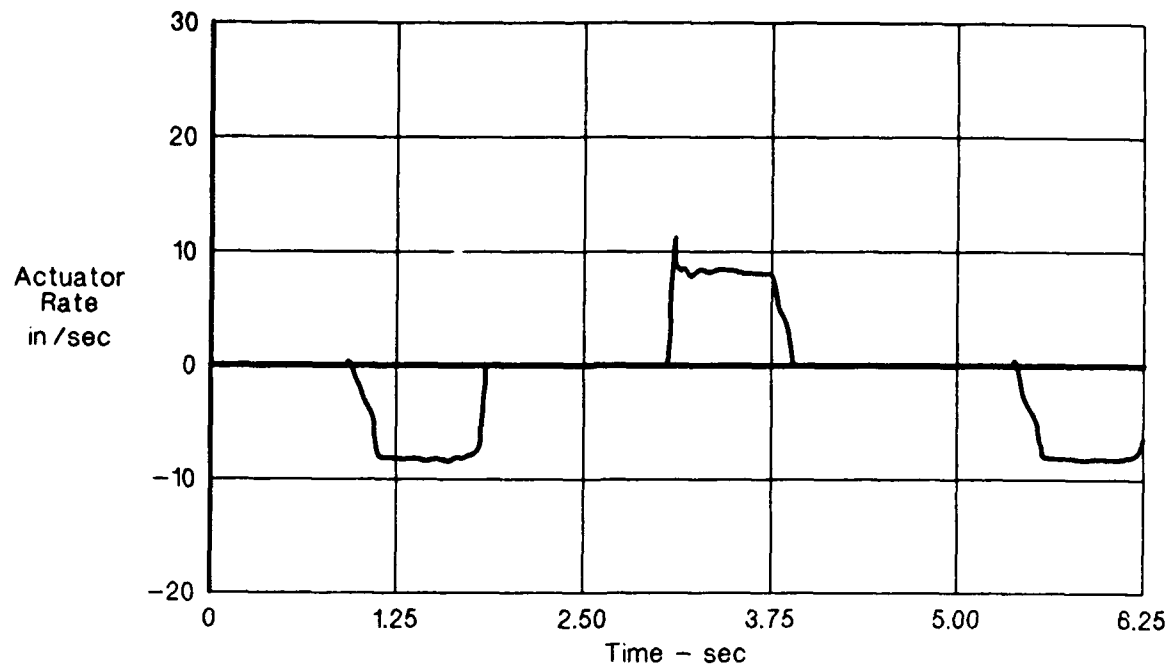


Figure 309
No-Load Step Response - 8,000 psi - Full Up
Configuration - Actuator Rate

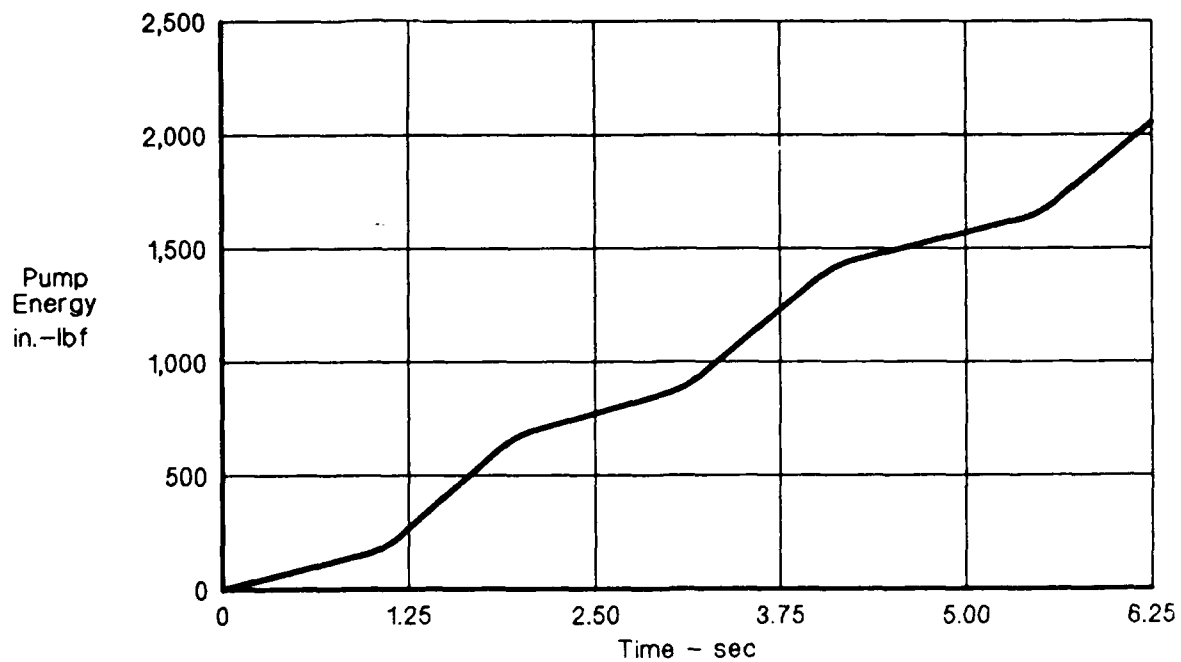


Figure 310
No-Load Step Response - 8,000 psi - Full Up
Configuration - Acurex Pump Energy

Figures 311 through 319 present test data for the conventional system. The actuator was cycled in the extend direction, then reversed to the retract direction, and back to the extend direction before the actuator was stopped. Figure 311 shows the pressure transducer output at the pump. After an initial dip in pressure to approximately 6,500 psi, the output pressure fluctuated between 8,200 and 7,500 psi.

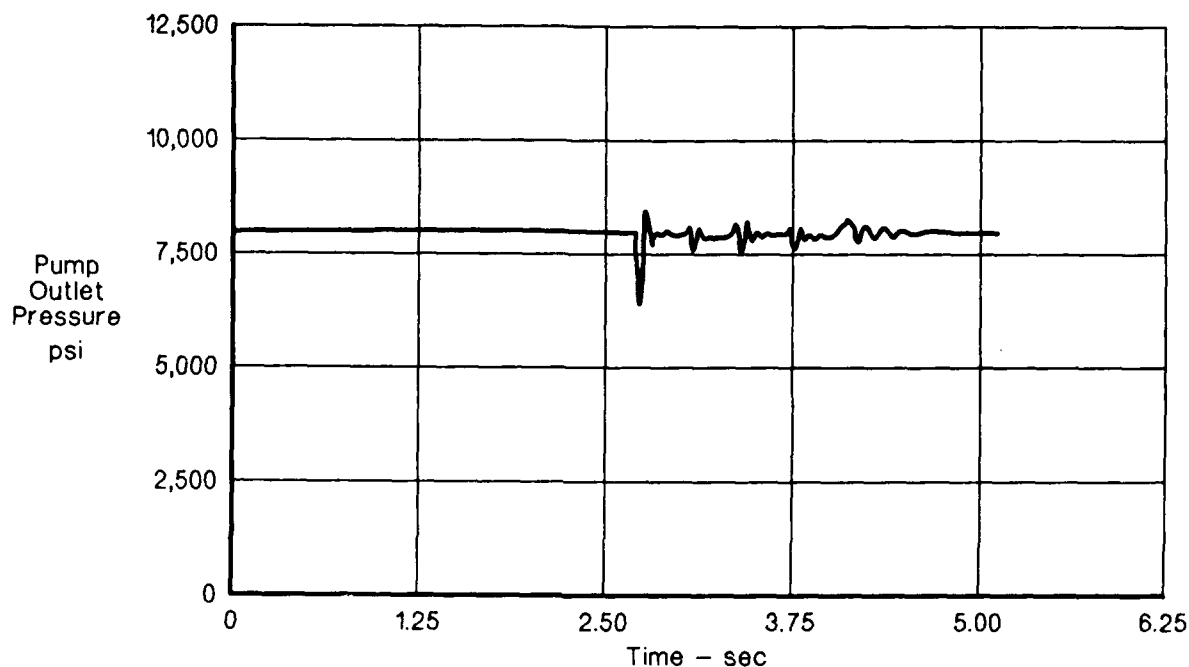


Figure 311
No-Load Step Response - 8,000 psi - Conventional -
Pump Outlet Pressure

Figure 312 presents the system 1 pressure transducer output located immediately upstream of the actuator. The pressure varied between 4,000 and 2,000 psi as a function of the actuator direction of operation due to the unbalance of the system 1 piston. Figure 313 presents the return system 1 pressure transducer output located immediately downstream of the actuator. The initial pressure spike (2,800 psi), was caused by dumping intermediate cylinder pressure to the return system upon valve opening. The steady-state running pressures were approximately 1,800 and 1,200 psi, and depended upon direction of actuator motion (due again to the unbalanced piston area).

Figure 314 shows pressure data recorded immediately upstream of the system 2 (balanced), side of the actuator. The steady-state running pressures were approximately 2,500 and 2,000 psi. These pressures should be the same in both directions for identified rates. However, the rates in each

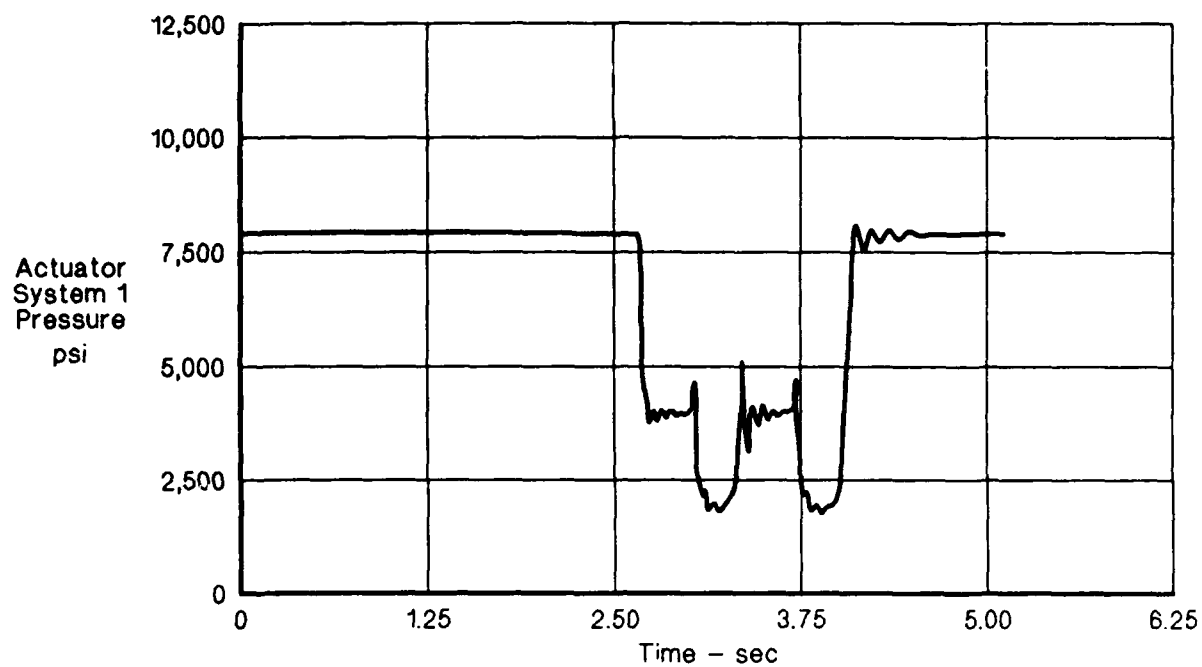


Figure 312
No-Load Step Response - 8,000 psi
Conventional - System 1 Pressure

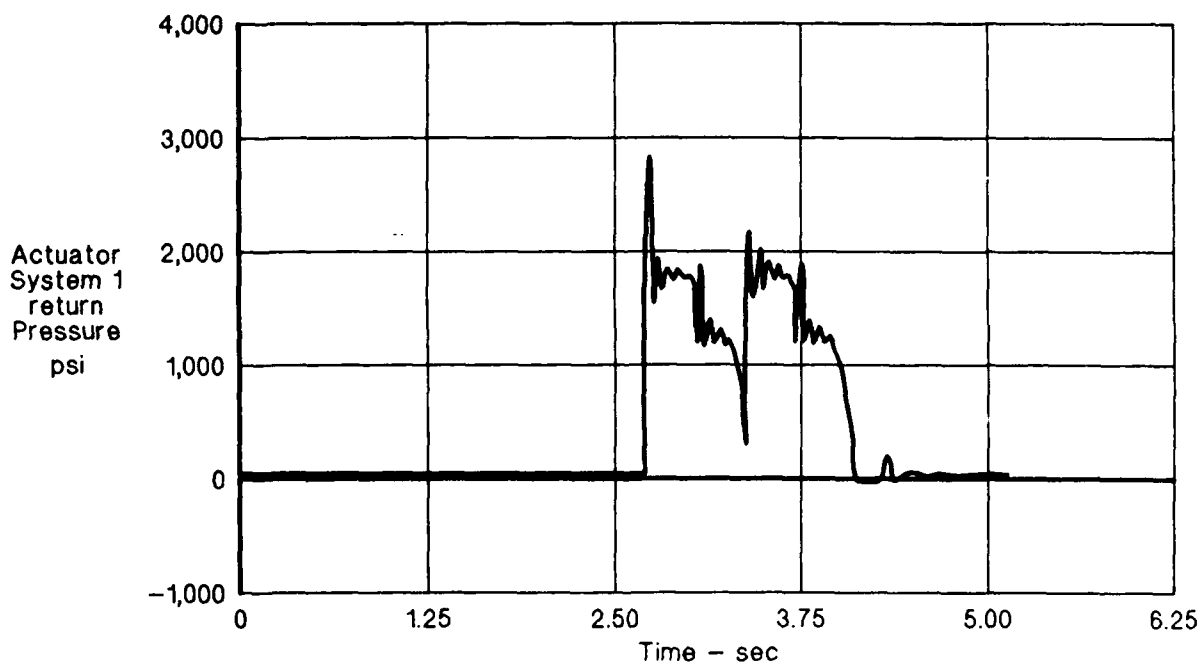


Figure 313
No-Load Step Response - 8,000 psi
Conventional - System 1 Return Pressure

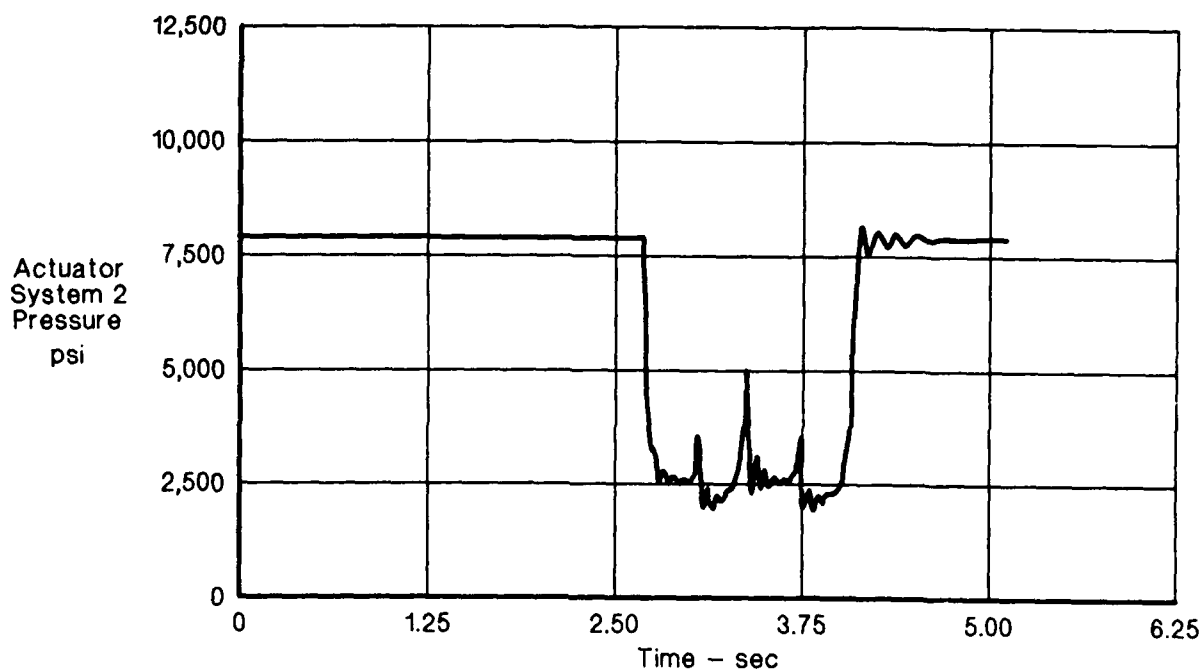


Figure 314
No-Load Step Response - 8,000 psi
Conventional - System 2 Pressure

direction of operation were different, extend being somewhat faster. This explained the variation. The pressure spikes noted, superimposed on the steady-state pressures, were due to water hammer associated with the slowing down of the fluid when the actuator was reversing direction. Figure 315 presents the system 2 return pressure data immediately downstream of the actuator. The initial spike due to dumping of cylinder pressure was 2,800 psi. The running pressures were 1,800 and 1,200 psi even with the system 2 balanced piston, because only one restrictor was used downstream of the combined system 1 and 2 return flows to control actuator speed. Thus, the unbalanced side effects were "fed over" and affected the balanced side.

Figure 316 shows the combined system 1 and 2 return side flow during cycling. Figure 317 presents the actuator control valve position commanded in inches. The valve is "popped" hard over in one direction and then the other. Figure 318 presents the main ram position vs. time in response to valve input commands. Figure 319 presents the derived actuator rate in in/sec. for both directions of operation. The retract direction was slightly faster than extend (approximately 9.0 vs. 8.5 to 8.7 in/sec.).

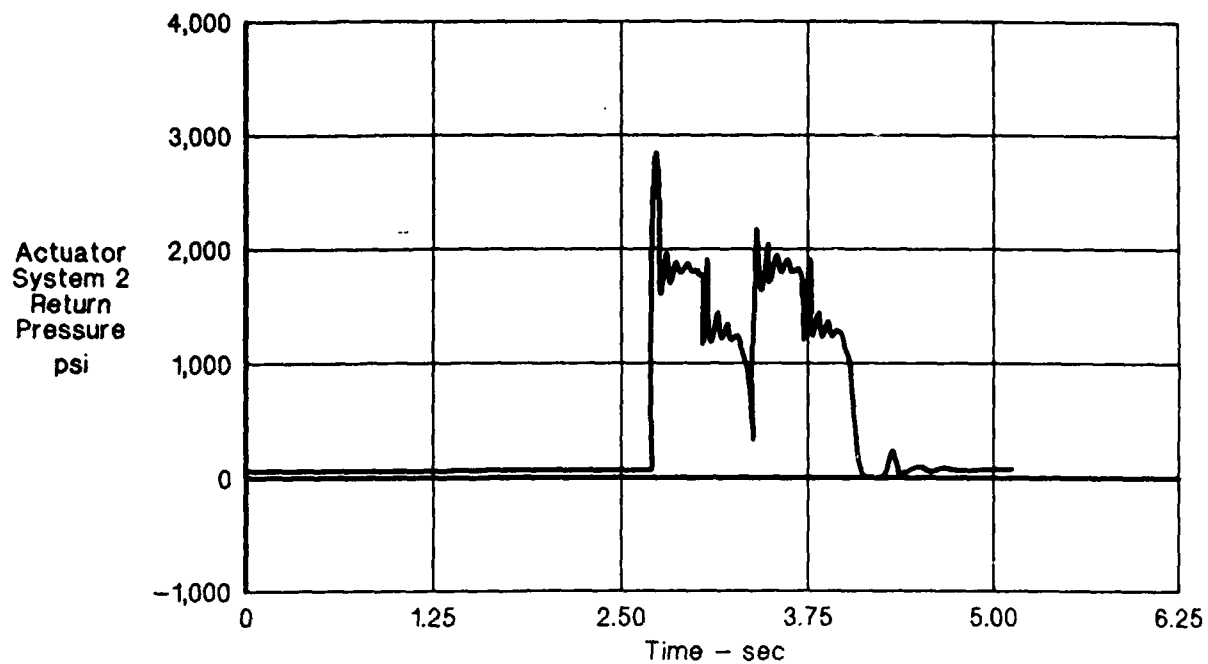


Figure 315
No-Load Step Response - 8,000 psi
Conventional - System 2 Return Pressure

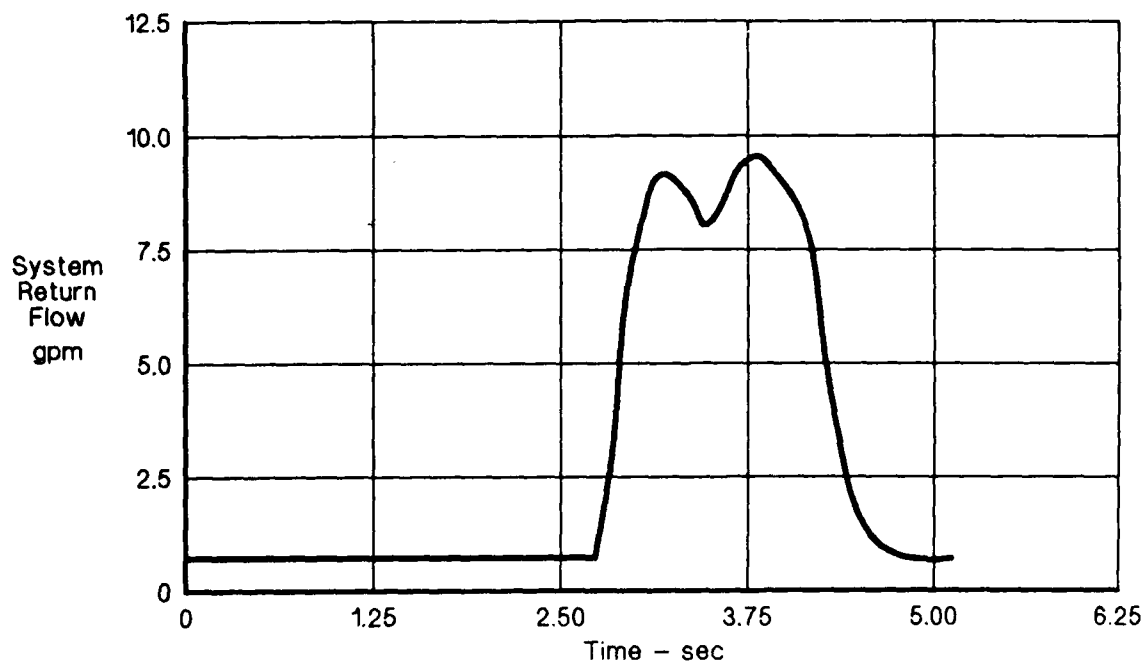


Figure 316
No-Load Step Response - 8,000 psi
Conventional - System Return Flow

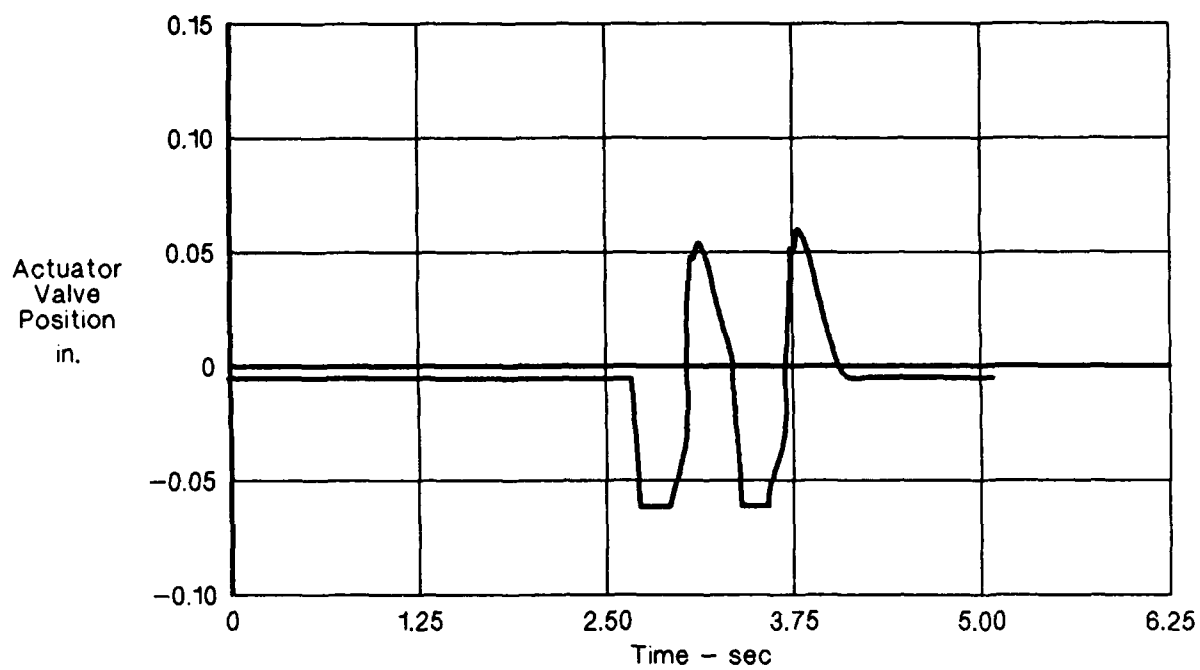


Figure 317
No-Load Step Response - 8,000 psi - Conventional -
Actuator Valve Position

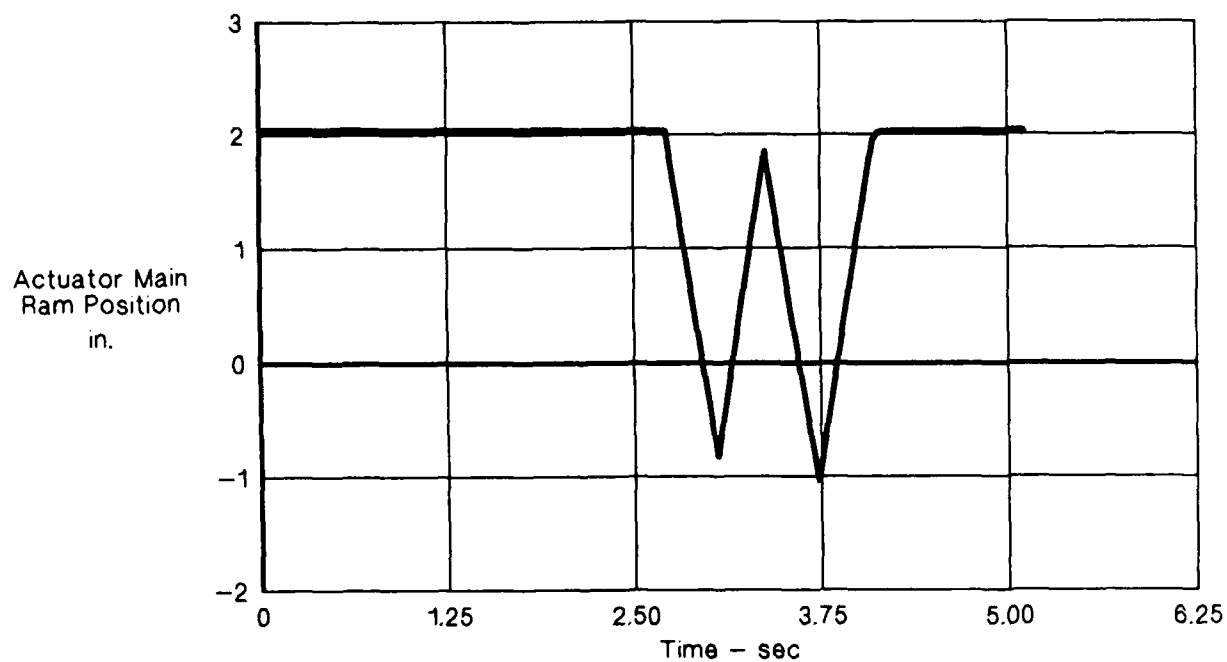


Figure 318
No-Load Step Response - 8,000 psi - Conventional -
Main Ram Position

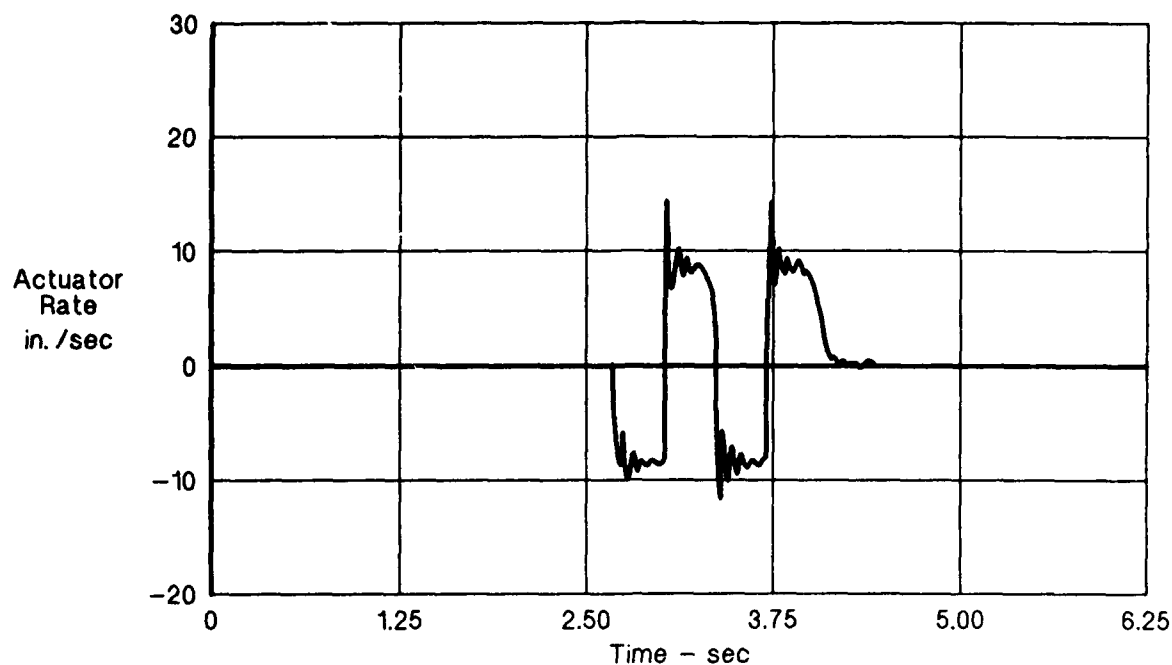


Figure 319
No-Load Step Response - 8,000 psi - Conventional -
Actuator Rate

c. Variable Pressure Pump - During a MCAIR study of an F-18 mission profile, it was shown that the flight controls and hydraulic utility function did not require full system operating pressure at all times. Data revealed that 93.7 percent of the mission could be accomplished at 1,000 psi, vs. 3,000 psi. With the advent of relaxed static stability in the current generation of aircraft, others feel that it would be closer to 75 percent of the time. The key is that a variable pressure pump does not always operate at its peak pressure. Because it is not always at its peak pressure, there is a reduction in both energy and heat rejection. Less heat rejection means that weight could be reduced by using a smaller heat exchanger. Tests were run with an Abex variable pressure "Smart Pump" to see exactly how much energy and weight could be saved.

As stated above, there are two primary benefits to variable pressure. The first is reduced energy consumption by the pump (less input horsepower), and the second is reduced heat rejection into the system, which yields a smaller system heat exchanger. Several tests were performed to verify and quantify these benefits. Another less measurable but potentially significant

benefit, is increased seal life, since 75 percent or more of the system's life is spent at 3,000 psi as opposed to 8,000 psi.

Reduced Energy Consumption - To make a better comparison of energy consumption between a constant 8,000 psi system and a 3,000 to 8,000 psi variable pressure system over a simulated mission profile, the actuator was put through the endurance duty cycle shown in Figure 320. The 72-second cycle was designed so the variable pressure pump would operate at 3,000 psi 75 percent of the time, and at some elevated pressure during the remaining 25 percent of the time. Pump torque was recorded over each 72-second cycle at constant 8,000 psi, and with variable pressure. The torque was converted to energy, and as shown in Figure 321, the 75 percent 3,000 psi system used almost half as much energy with variable pressure compared to constant 8,000 psi. The average horsepower over one duty cycle was found, and as shown in Figure 321, by using a variable pressure system, the average horsepower could be reduced from 19.86 hp to 10.38 hp, a reduction of 48 percent.

Reduced System Heat Rejection - A number of pump heat rejection tests were performed on both the primary and the backup smart pump. Pump torque was measured during speed sweeps from 1,500 to 4,500 rpm at constant pressure (either 3,000, 5,500, or 8,000 psi) and temperature. Figures 322 through 325 show plots of pump case drain flow and pump input torque at 8,000 psi and 3,000 psi with an inlet temperature of 160°F. The torque and case drain flow spike which occurred at 2,200 rpm at 8,000 psi and 1,950 rpm at 3,000 psi was present on both smart pumps, as well as the 15 gpm FWFRHS constant pressure pump. There was no change in pump outlet pressure at this spike, however.

Torque data was converted to horsepower to get a measure of energy consumption. Pump horsepower vs. speed (rpm) plots for each temperature are presented in Figures 326 and 327. These plots show that operating at 3,000 psi resulted in a 45 percent reduction in energy consumption over 8,000 psi operation (6.1 hp vs. 11.1 hp at 4,000 rpm). This occurred despite the fact that there was little decrease in case drain flow when operating at 3,000 psi vs. 8,000 psi. In many cases, the case drain flow was higher (see Figures 322 and 323). The case drain flow phenomenon was EHV related (EHV control

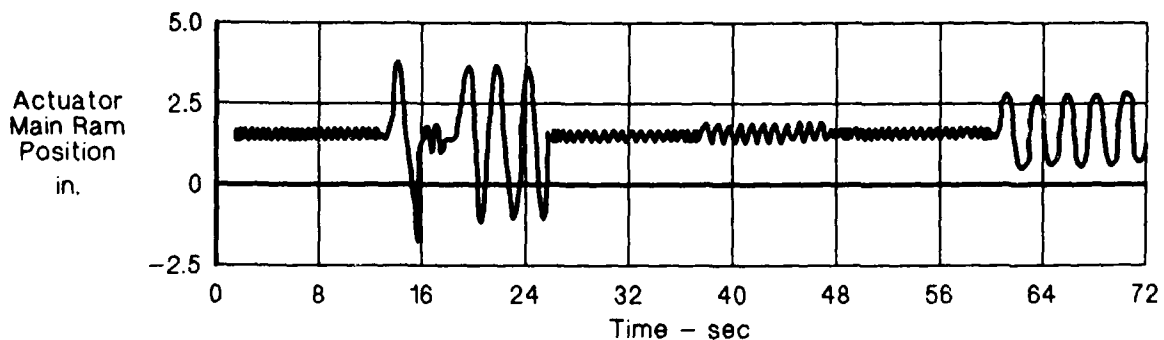


Figure 320
Endurance Test Duty Cycle

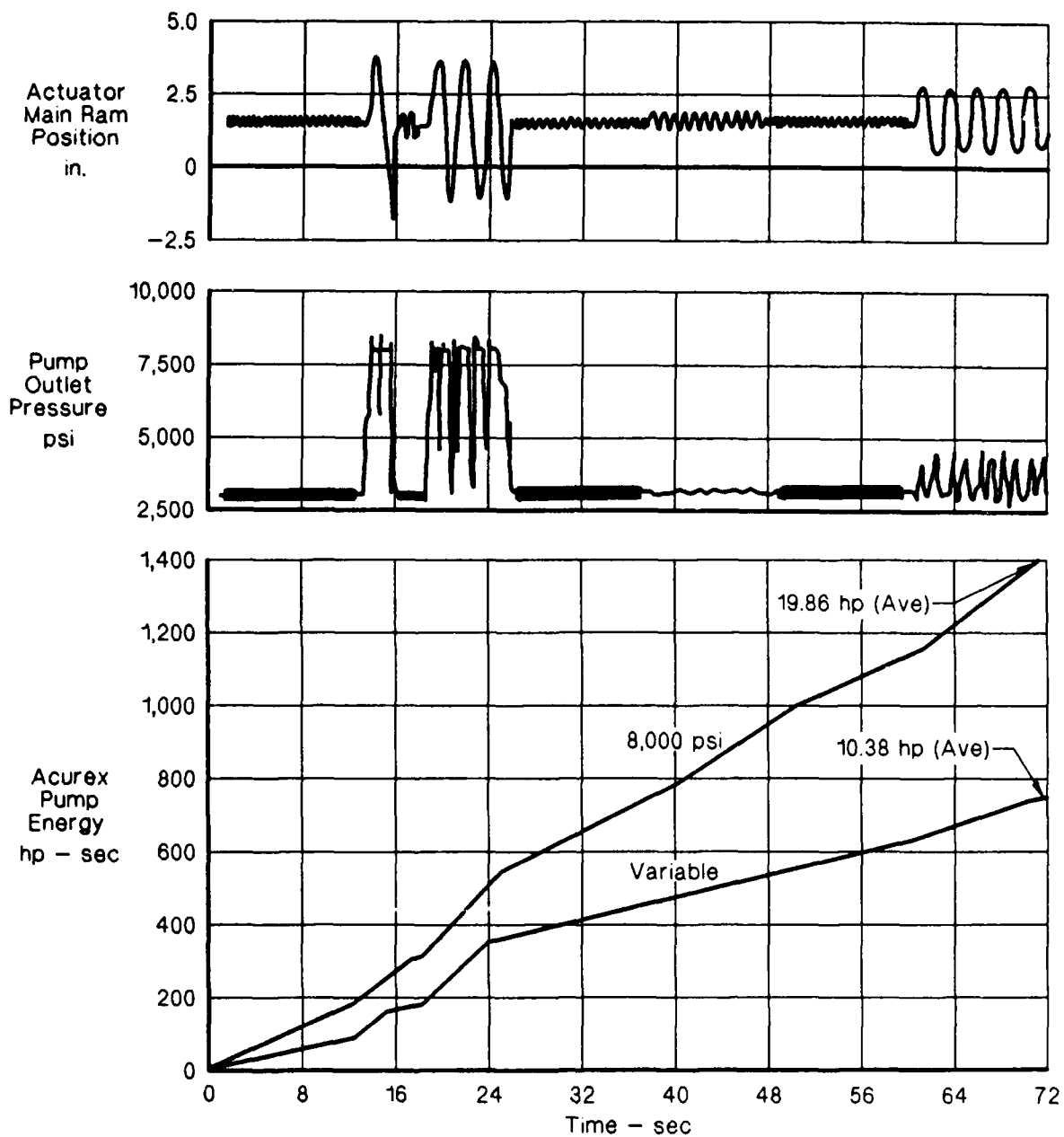


Figure 321
Variable Pressure Energy Comparison

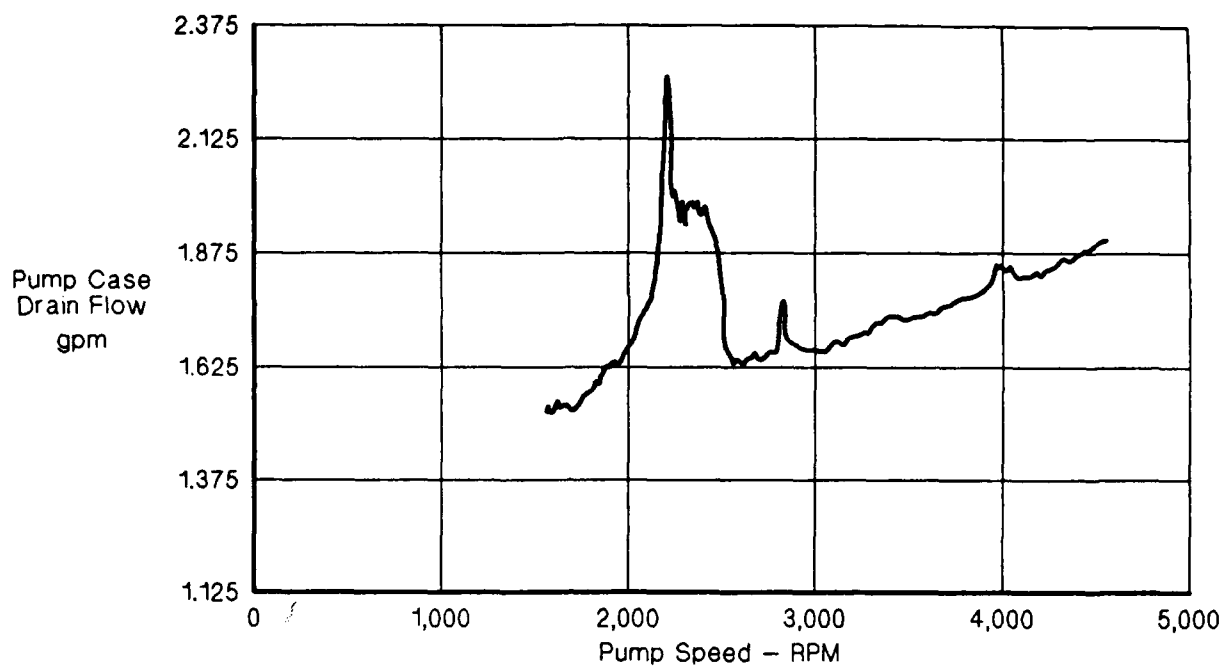


Figure 322
8,000 psi Case Drain Flow vs. Pump Speed

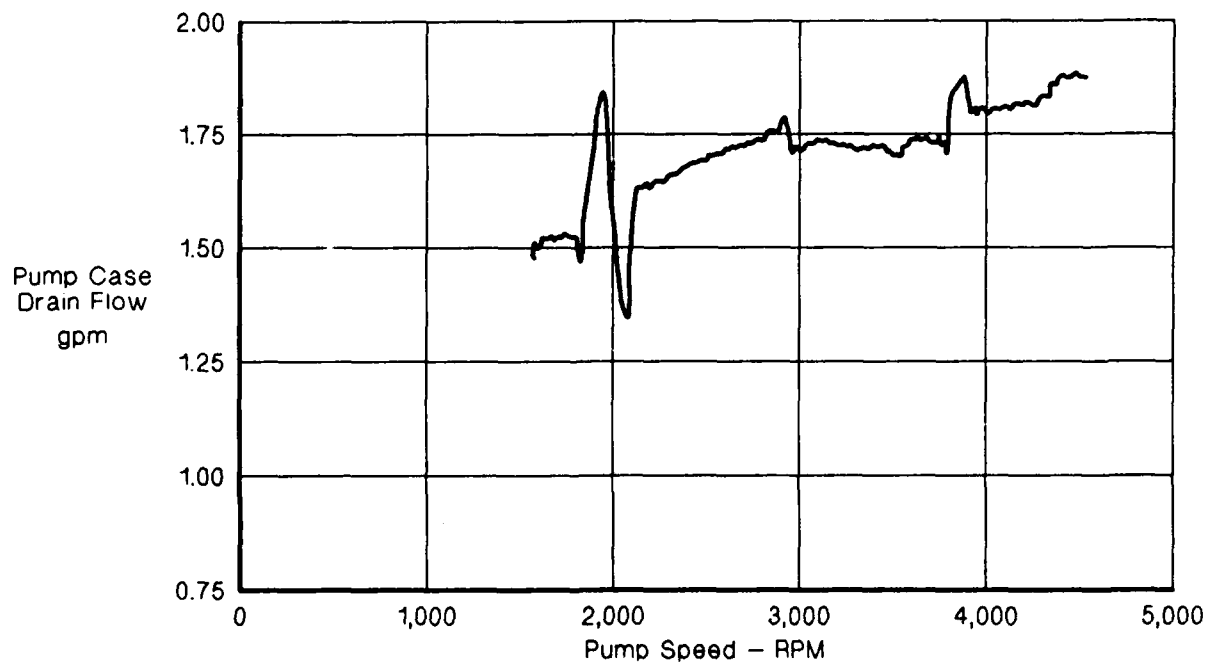


Figure 323
3,000 psi Case Drain Flow vs. Pump Speed

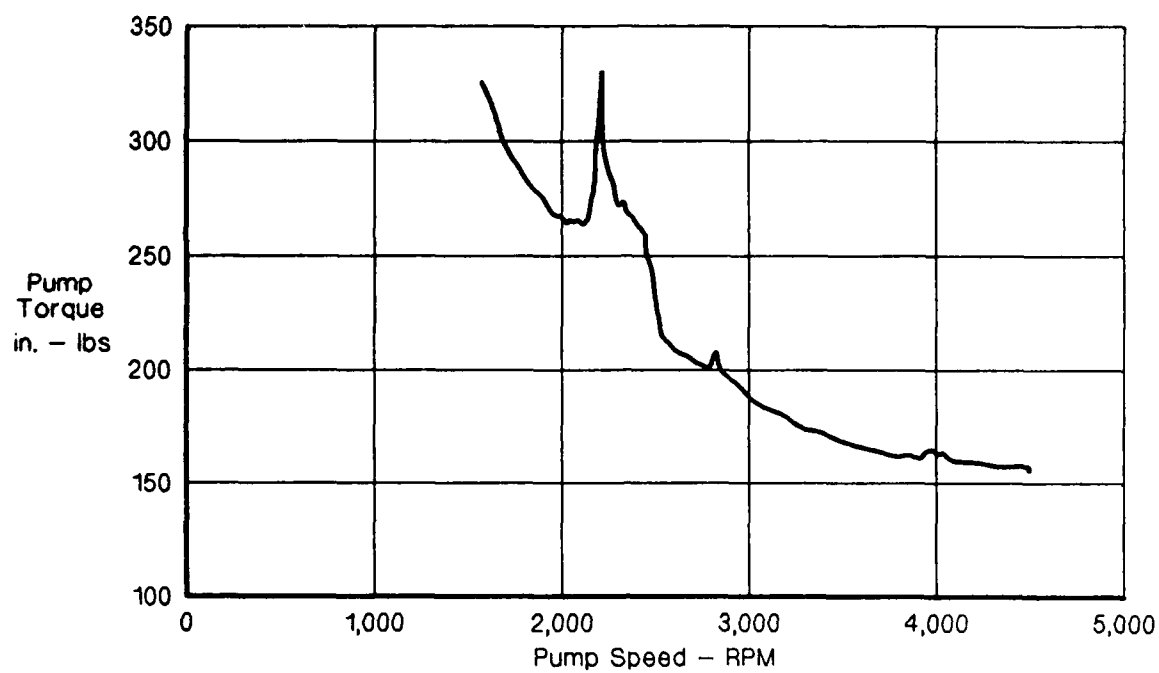


Figure 324
8,000 psi Pump Torque vs. RPM

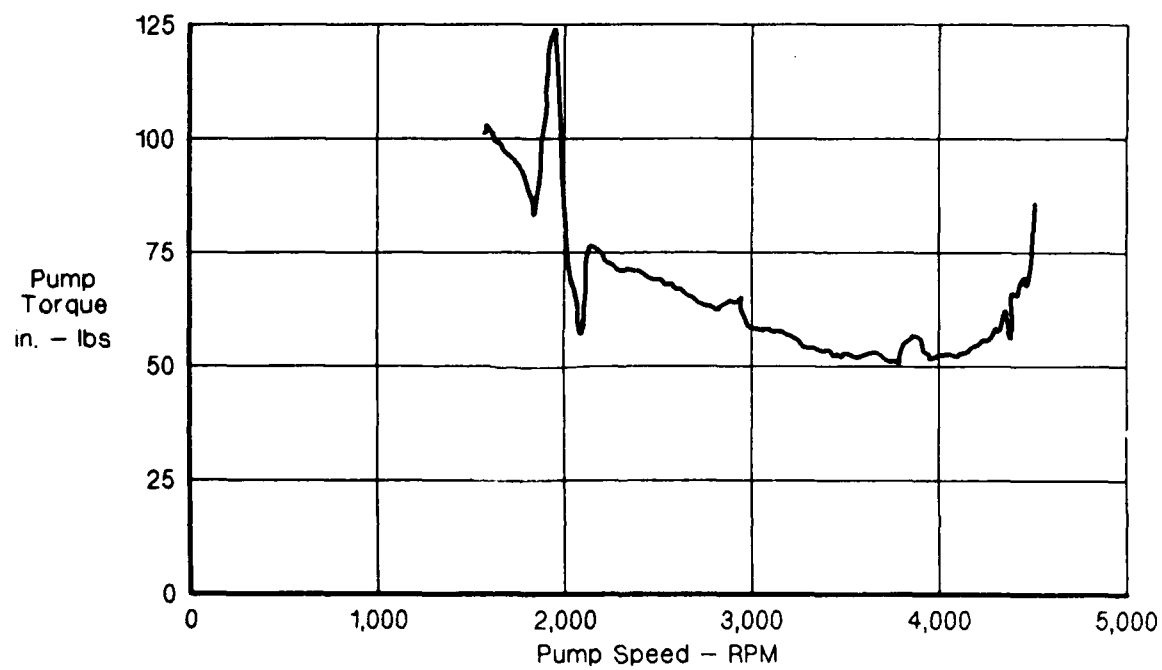


Figure 325
3,000 psi Pump Torque vs. RPM

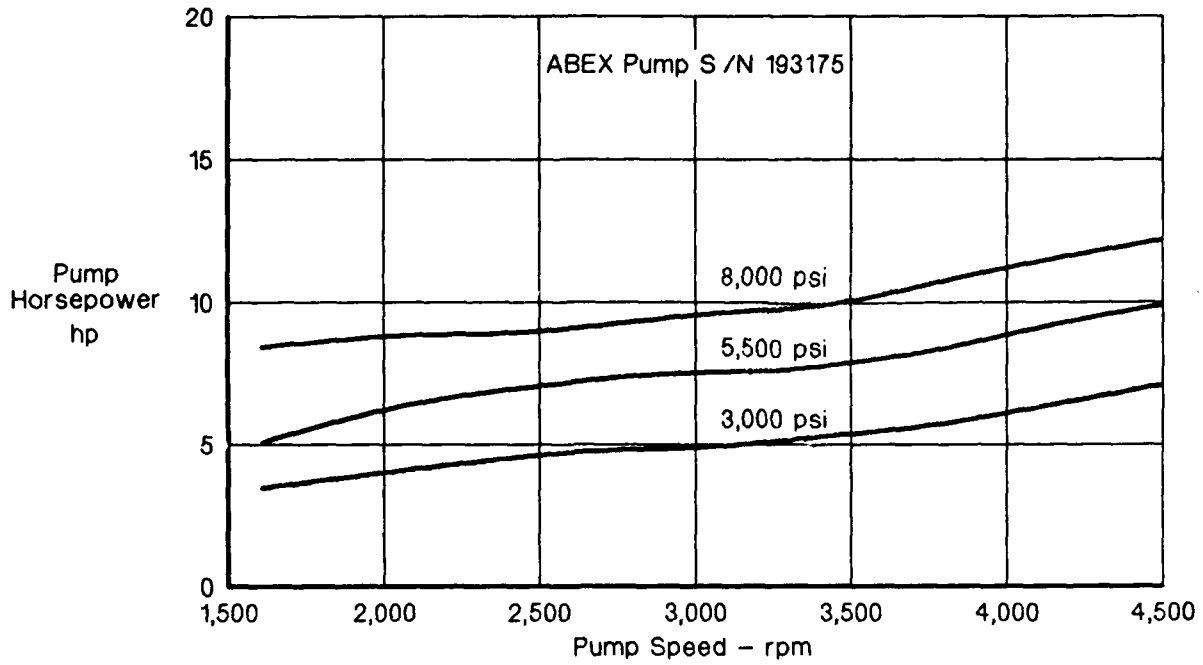


Figure 326
Pump Heat Rejection - At 160°F Inlet Temperature

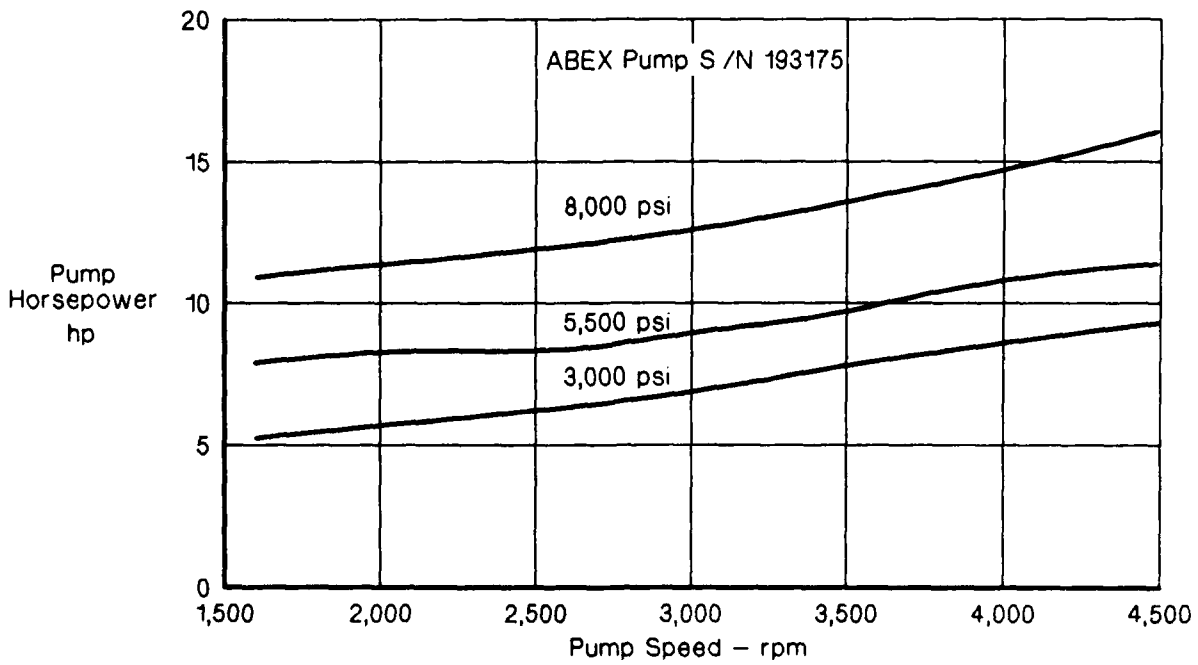


Figure 327
Pump Heat Rejection - At 275°F Case Drain Temperature

return flow was dumped into the case drain), since it consistently occurred on the two smart pumps, but not on the FWFRHS constant pressure pump.

To find out how much the heat exchanger weight could be reduced, two systems were analyzed by the MCAIR Thermodynamics Department: a constant 8,000 psi (baseline) system, and a variable pressure system on the endurance test duty cycle. As shown in Figure 328, temperatures were taken throughout

	Variable Pressure Temperatures (°F)	Constant Pressure Temperatures (°F)
Pump _{out}	106.2	124.7
Pump _{in}	87.3	106.4
Pump _{cd}	139.9	171.2
HX _{in}	126.4	160.2
HX _{out}	86.8	108.7
Water _{in}	40.0	40.0

Figure 328
LECHT Demonstration System
Steady-State Temperatures

both systems while they were running. Because temperatures were higher in the baseline system, the baseline heat exchanger was analytically resized so that its case drain temperature declined from 171.2°F to the variable pressure's case drain temperature of 139.9°F. This showed how much larger the baseline system's heat exchanger must be in order to obtain the same temperature as the variable pressure system. The weight and volume of the heat exchanger can be seen as a function of the water flow in Figures 329 and 330. As the water flow increased, the weight of the heat exchanger decreased. The amount of the weight which can be saved, can be clearly seen on a graph of the weight of the heat exchanger vs. the water flow rate (Figure 331). For the same water flow rate, the modified system weighed about 2.25 times that of the baseline system. This meant that for a constant pressure hydraulic system to run at the same temperature as a variable pressure system, the heat exchanger would have to be over two times as large.

There were also a few negatives brought on by variable pressure. System fatigue was more severe and must be accommodated for in the design. Actuator frequency response at low amplitudes was reduced as expected, with an acceptable performance level and an increase in system complexity. But the performance was acceptable. There was also an increase in system complexity.

Water Flow (lb/min)	Weight (lb)	Vol (in. ³)
50	0.75	12.0
40	0.79	12.0
30	0.85	13.0
20	0.99	15.0
10	1.20	20.0
5.75	2.21	38.0

Figure 329
Baseline System Heat Exchanger Weights

Water Flow (lb/min)	Weight (lb)	Vol (in. ³)
50	1.56	27.0
40	1.65	28.0
30	1.81	31.0
20	2.17	38.0
10	3.09	57.0
7.86	4.75	91.0

Figure 330
Modified System Heat Exchanger Weights

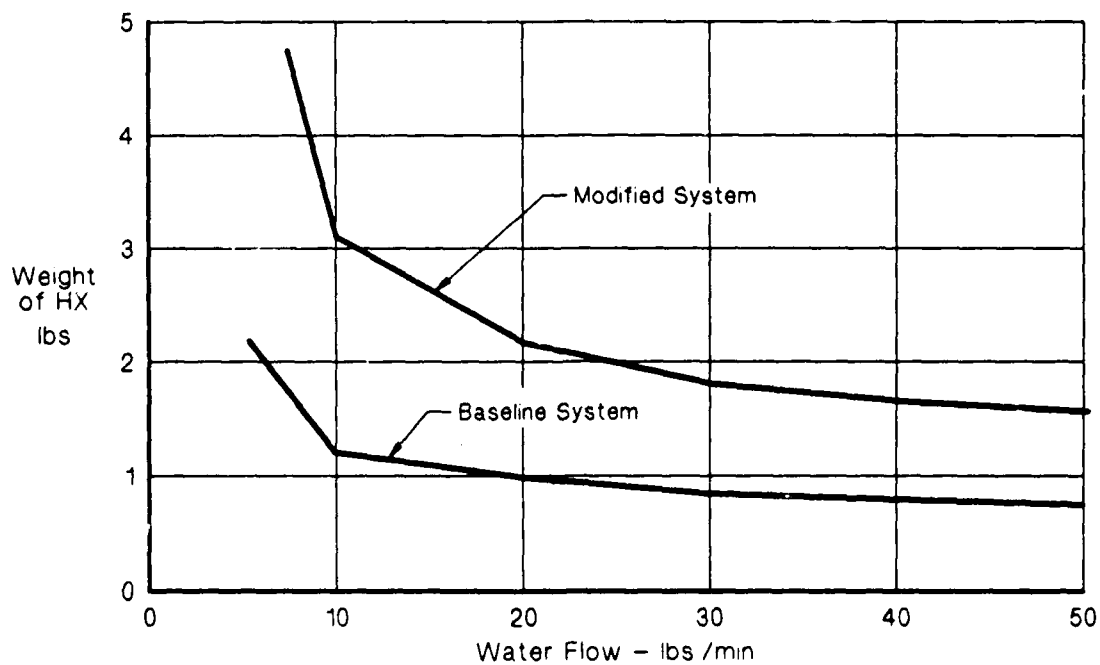


Figure 331
Weight of Heat Exchanger vs. Water Flow

d. General Pump Performance - To evaluate the pump operating characteristics, tests were run to measure pump rated flow, hysteresis and linearity, and frequency response. Rated flow was checked at both 3,000 and 8,000 psi system pressure. The bypass valve was opened to drop pump pressure to 7,850 (or 2,850 psi) with the pump running at 4,000 rpm, and outlet flow was recorded. At both 3,000 and 8,000 psi, the maximum flow rate was 15.5 gpm. Hysteresis and linearity tests were performed next. The Abex pump was nonlinear and varied as much as 250 psi at times (see Figure 332). This was

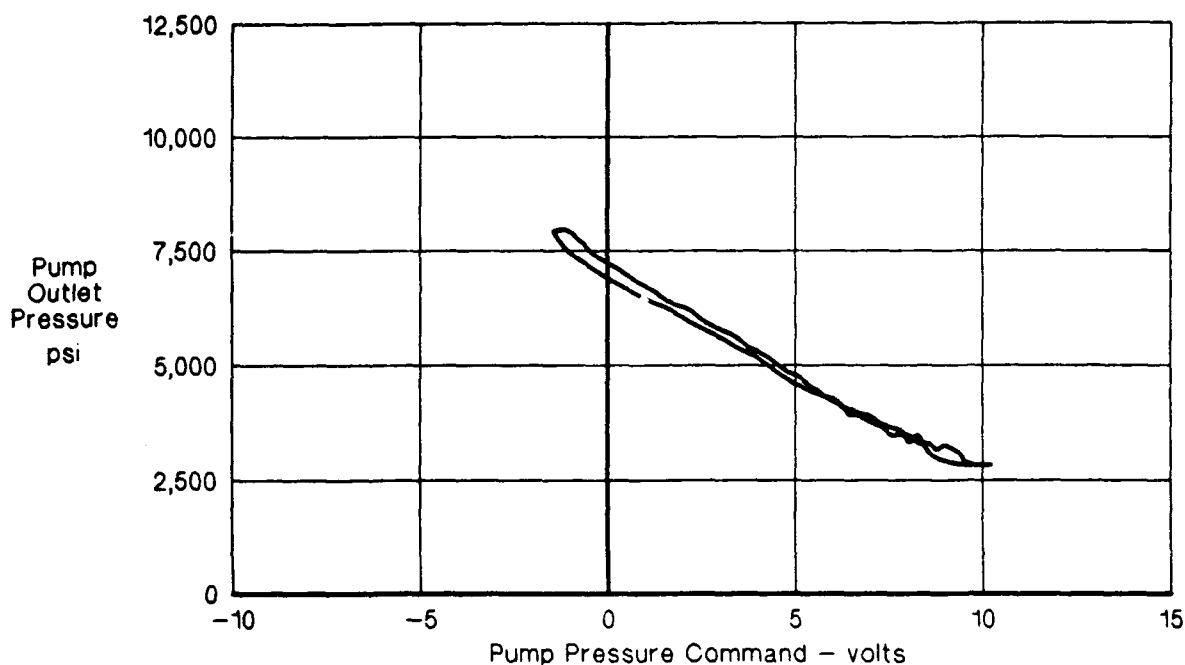


Figure 332
Abex Pump Hysteresis at 4,000 RPM, 8,000 psi

not unexpected since the Abex design was open loop. Frequency response data which is negatively affected by being an open loop design is presented in the form of a Bode plot in Figure 333.

e. General Actuator Performance - Several other tests were performed on the Bertea stabilator actuator to verify its performance, including hysteresis, threshold, no-load frequency response and 40 percent loaded frequency response. No-load valve reversals and 100 percent loaded step responses were discussed in detail in Section 5.2.1.b.

Baseline Configuration Performance - Performance testing of the stabilator servoactuator began with the baseline configuration. This consisted of the "Smart Pump" operating at constant pressures and line-to-line valve installed in the manifold. In addition, flow augmentation devices were removed, and the load recovery valves were replaced by plugs made by MCAIR.

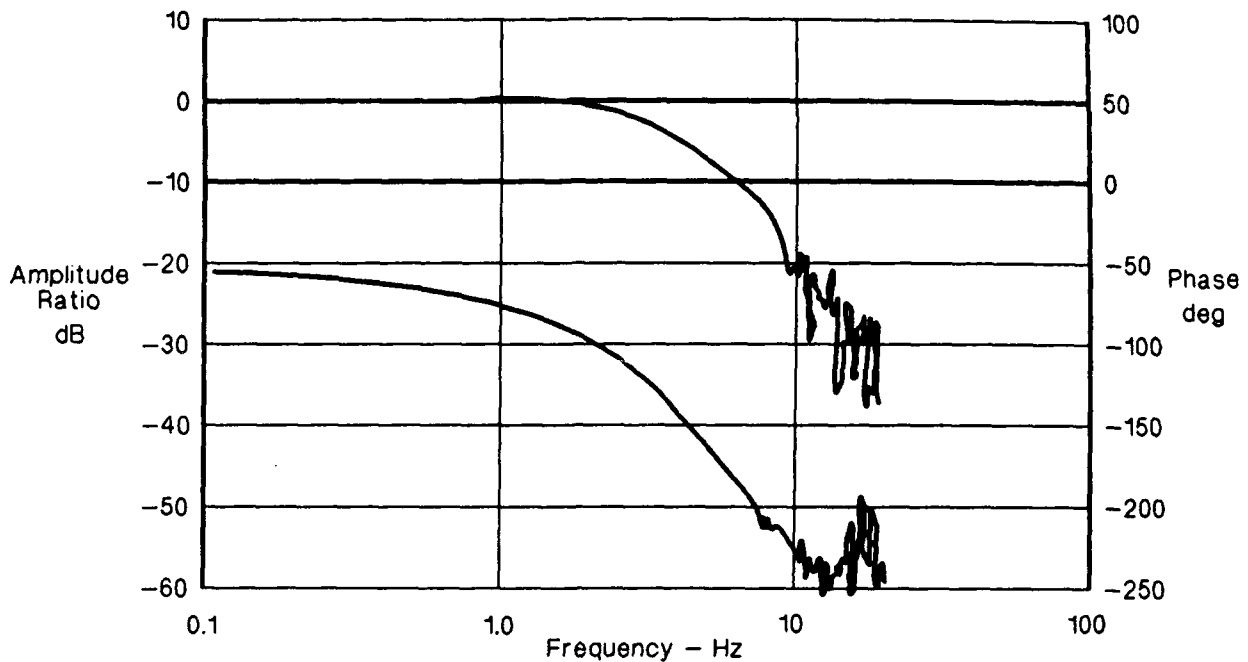


Figure 333
Variable Pressure Pump Frequency Response,
EHV Input Command to Outlet Pressure

Hysteresis was measured with respect to its electrical input command, at 0.01 Hz over its full operating stroke extending and retracting. As presented in Figure 334, there was nearly zero hysteresis over the full stroke. Similar results were obtained at 3,000 and 5,500 psi.

No-load frequency response tests were performed closed loop at each pressure at 160°F case drain fluid temperature with ± 1 percent and ± 10 percent main ram travel. Performance fell within F-15 SMTD specifications for amplitude and phase angle, as presented in Figures 335 and 336. These results were typical of the three pressures evaluated (8,000, 5,500 and 3,000 psi). In addition, the actuator was tested at 40 percent of stall load with ± 1 percent and ± 10 percent main ram travel. The results are shown in Figures 337 and 338.

Full up Configuration Performance - With the overlapped valve, load recovery valves and the jet pumps installed in the valve manifold, performance of the actuator showed no degradation. Section 5.2.1.b details the actuator's performance during no-load and loaded step responses. Figures 339 through 342 document the no-load and 40 percent stall load frequency response tests, performed at 3,000 and 8,000 psi with ± 2 percent and ± 10 percent stroke.

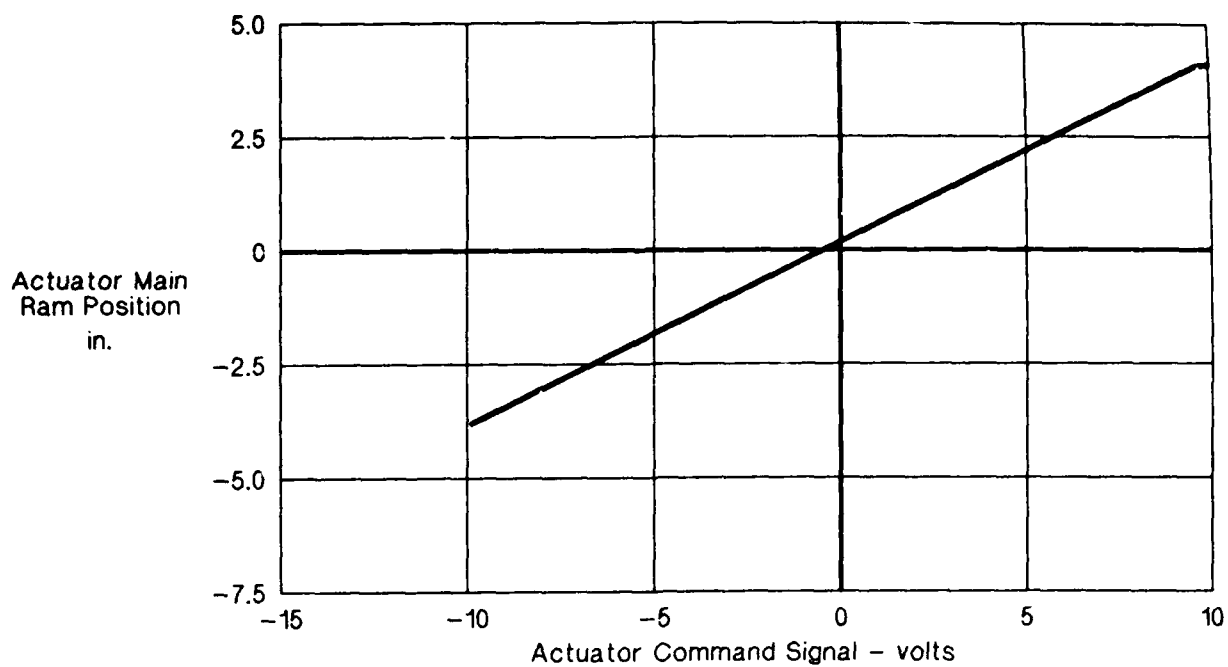


Figure 334
8,000 psi Stabilator Actuator Hysteresis

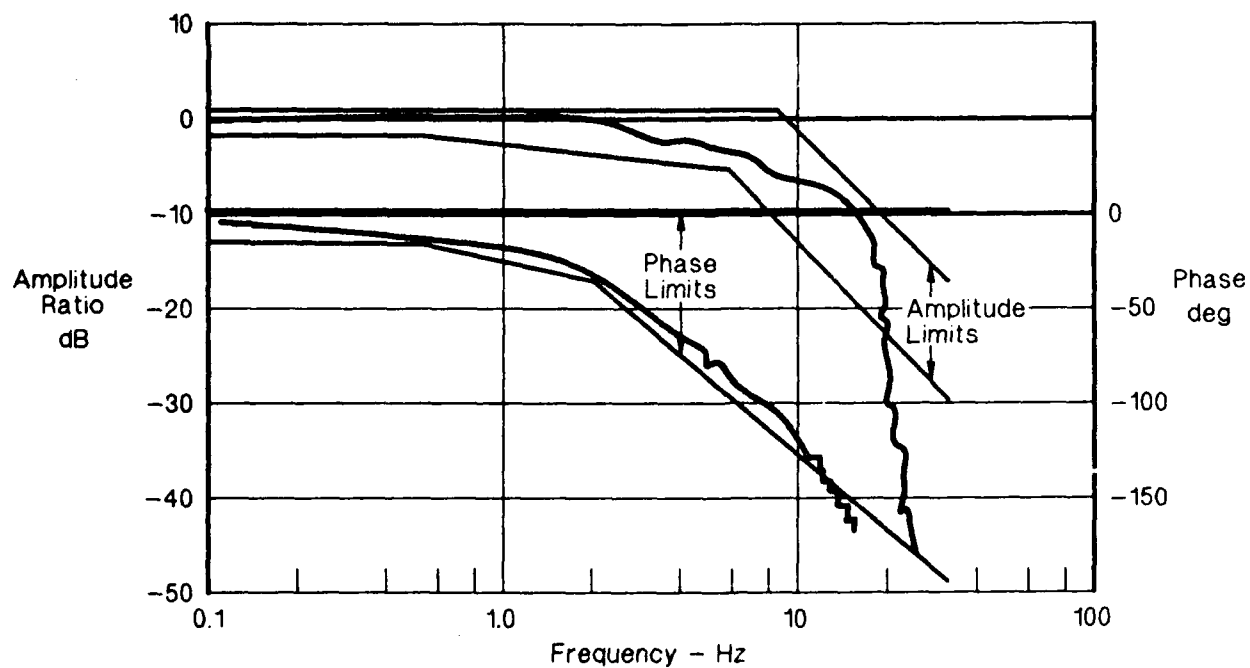


Figure 335
No-Load, 1% Main Ram Frequency Response
Baseline System - 8,000 psi

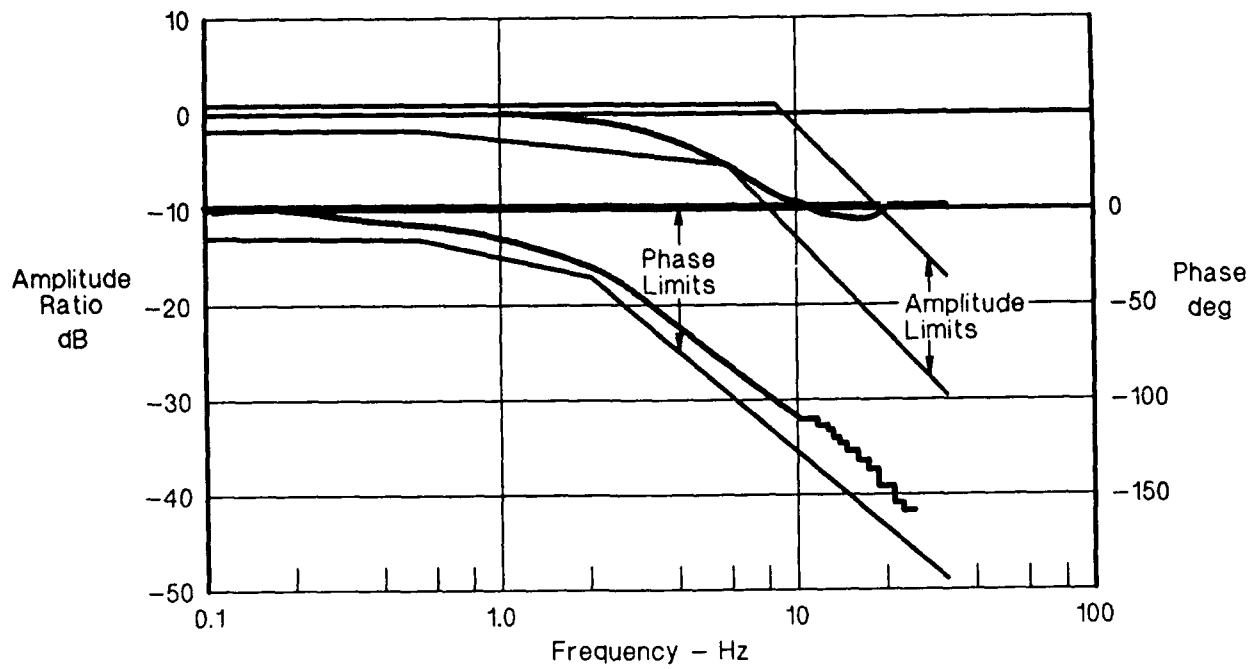


Figure 336
No-Load, 10% Main Ram Frequency Response
Baseline System - 8,000 psi

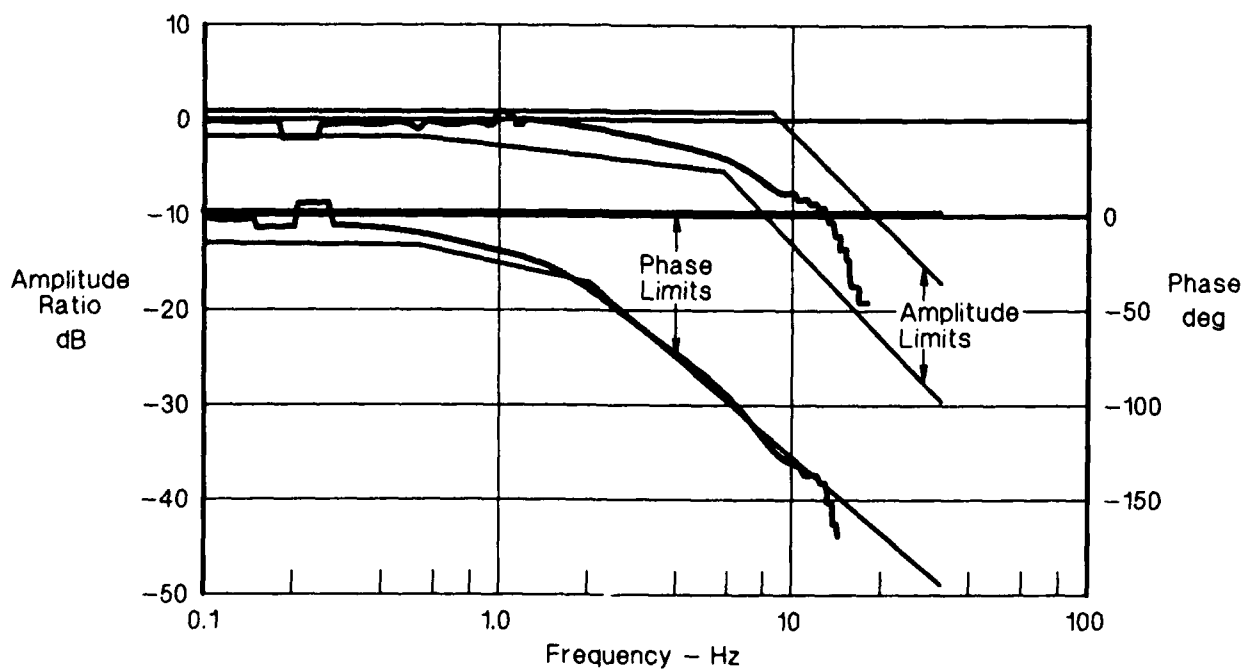


Figure 337
40% Stall Load, 1% Main Ram Frequency Response
Baseline System - 8,000 psi

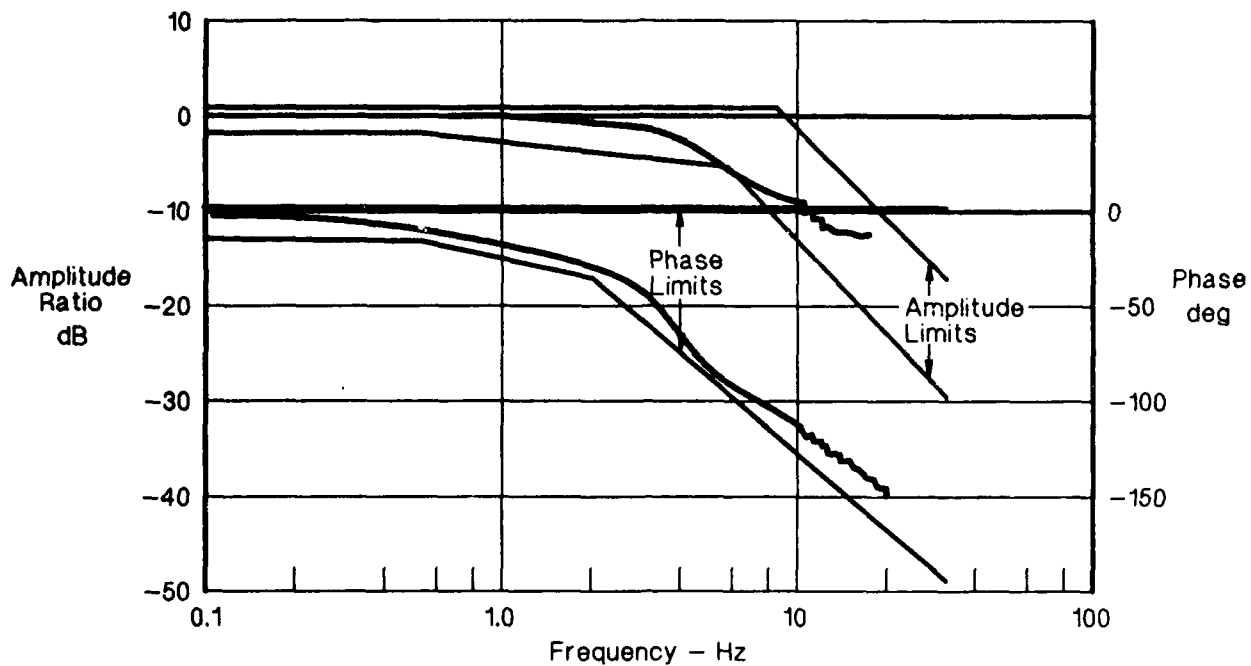


Figure 338
40% Stall Load, 10% Main Ram Frequency Response
Baseline System - 8,000 psi

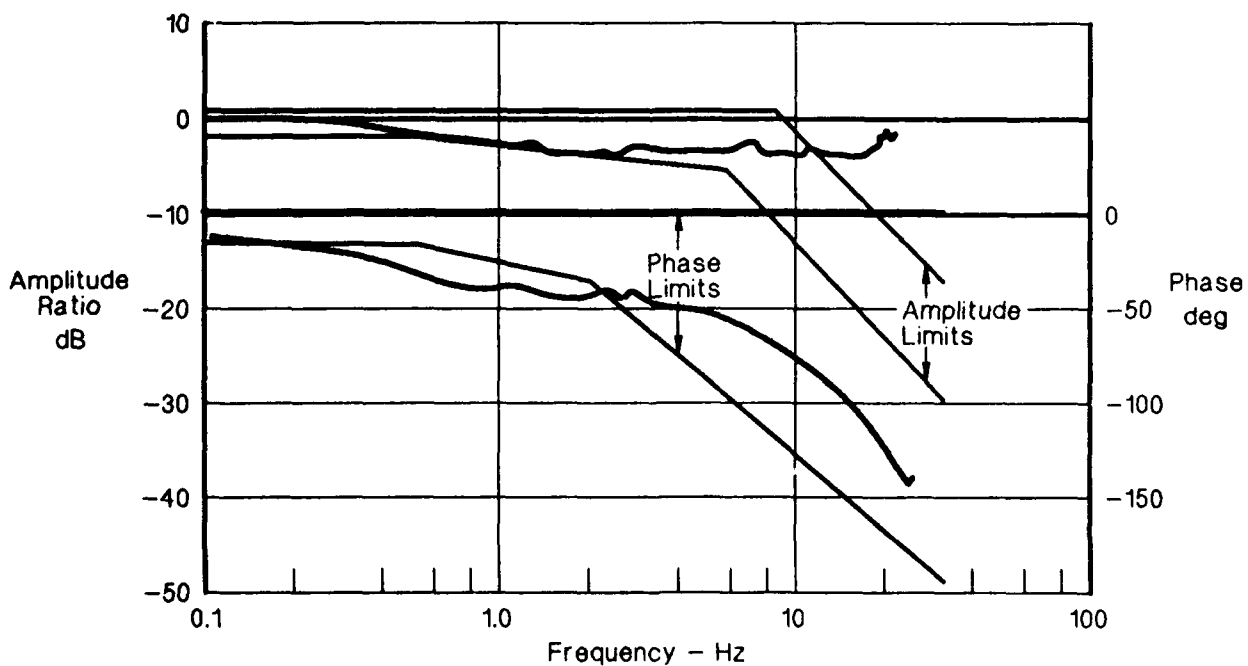


Figure 339
No-Load, 2% Main Ram Frequency Response
Full Up Configuration - 8,000 psi

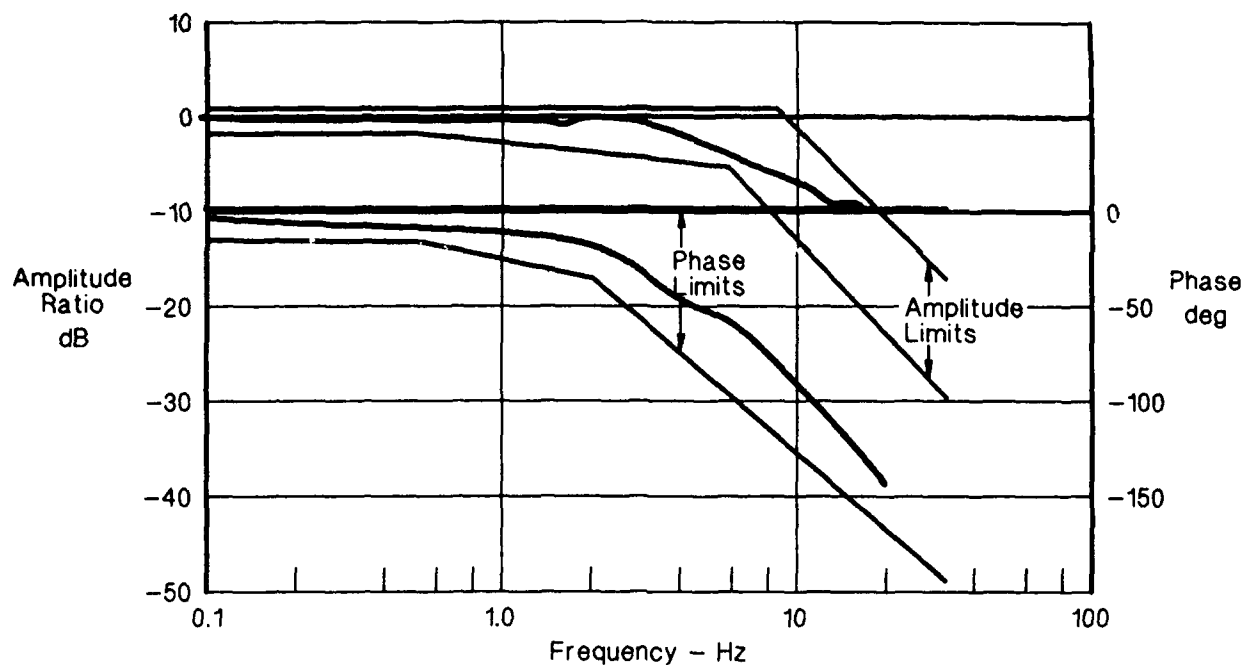


Figure 340
10% Main Ram Frequency Response
Full Up Configuration - 8,000 psi

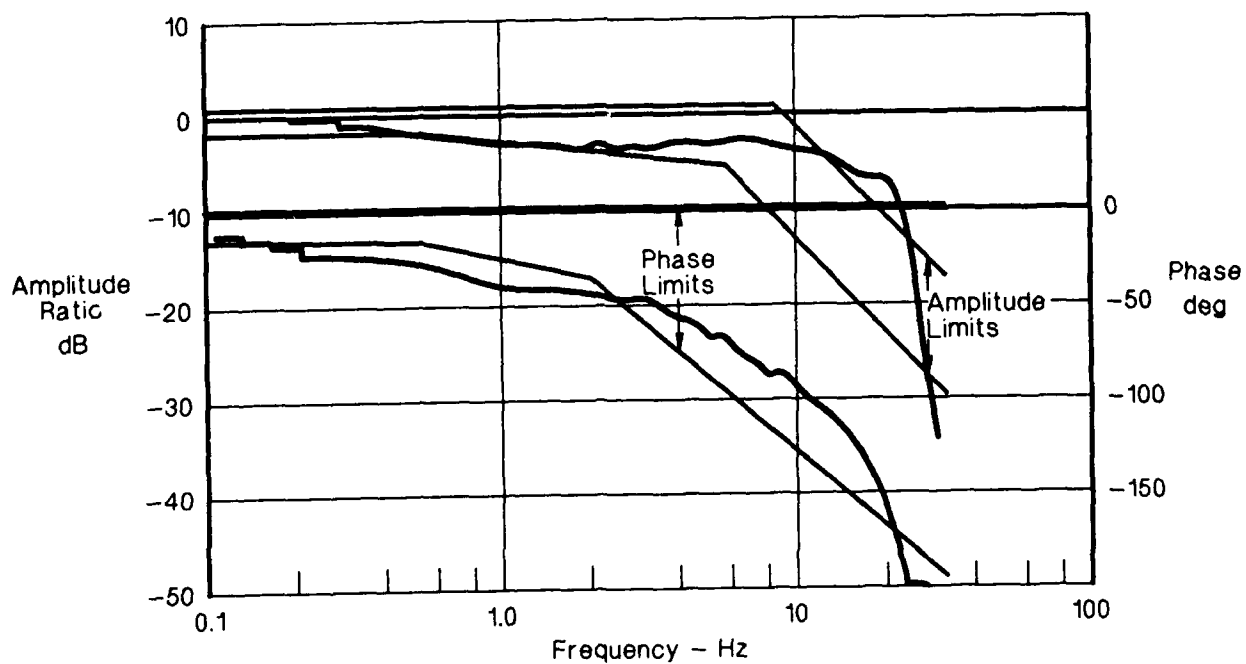


Figure 341
40% Stall Load, 2% Main Ram Frequency Response
Full Up Configuration - 8,000 psi

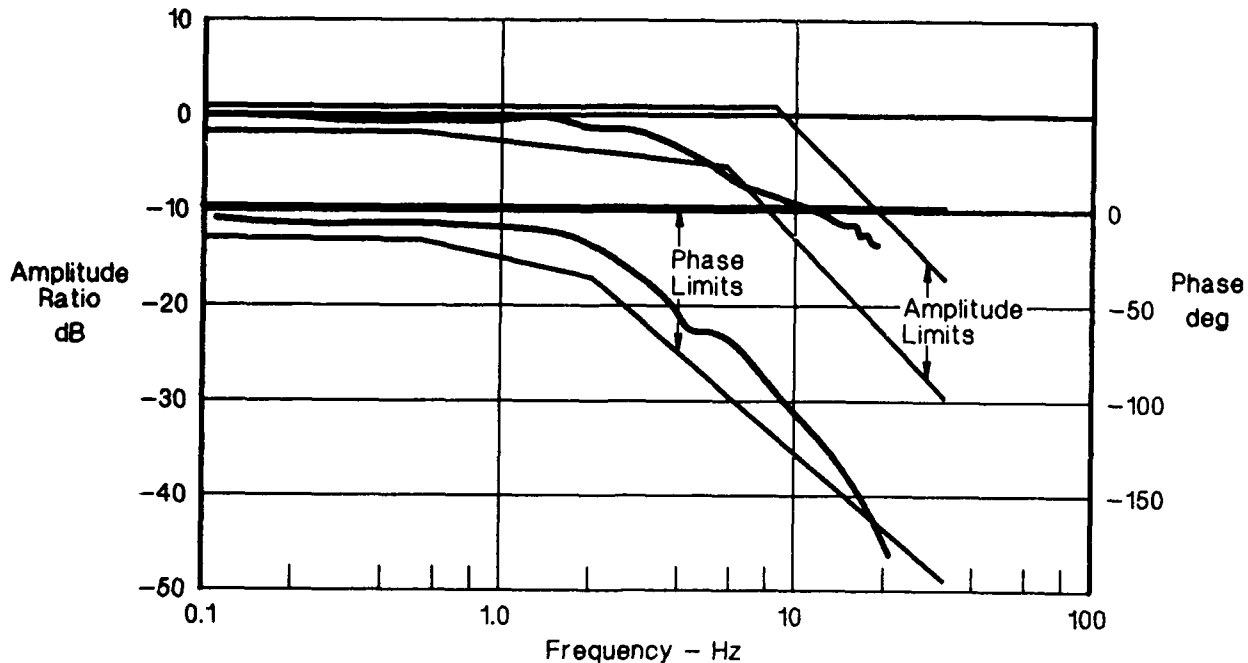


Figure 342
40% Stall Load, 10% Main Ram Frequency Response
Full Up Configuration - 8,000 psi

5.2.2 Endurance Test/System Performance - The endurance test, which was originally planned for 200 hours, was terminated at 139.4 hours because:

- o Problems with the MCV on the pump and the rod end of the load cylinder developed and supplier spare parts would not be available during the contract period of performance.
- o MCAIR and the Air Force were in concurrence to stop testing with a total test time of 245 hours in order to conclude the program without further contract extensions

a. Test Setup - The endurance test setup was the same as the performance evaluation test described in Section 5.1.2, see Figure 260. However, several items were added to control the actuator cycling and shut the system down in the event of component or system failure. An HP 9845B multiprogrammer microcomputer, along with a programmable frequency generator, was used to control the endurance test actuator cycling sequence and shutdown the pump drive if the following system related failures occurred:

- o System pressure fell below 2,000 psi
- o Pump case drain temperature exceeded 300°F
- o Reservoir fluid level fell below 10 percent of full

Figure 320 presents the 72-second cycling sequence for the endurance test. For each 72-second sequence, the stabilator servoactuator cycled 100 times. Therefore, after 139 hours of endurance testing, the stabilator piston accumulated 695,000 cycles, or the equivalent of about 16 miles of travel. The testing was completed in two blocks of 50 hours plus 39 hours, all with 100 percent external loads applied, as shown in Figure 343.

Block	Elapsed Time (hr)	Load on Actuators (percent)
1	50	100
2	50	100
3	39	100

Figure 343
Endurance Testing Loads

Between each 50-hour test block, the 72-second sequence was run at constant 8,000 psi pressure and variable 3,000 to 8,000 psi pressure to allow a comparison of the energy required. The sequence was designed so the system would operate at 3,000 psi for 75 percent of the time, which is what recent Air Force data revealed for future unstable, fly-by-wire aircraft.

b. Event History - The endurance test, along with the component acceptance/performance test, is summarized chronologically, in Figure 344. The following paragraphs present a detailed discussion of the endurance test events for each component, including a summary of the endurance test seal experience.

(1) Pump Experience - The Abex 15 gpm pump generally performed well, with the exception of the EHV which controlled the pressure setting. Three separate incidents required replacement of the EHV. The first occurred very early in the performance testing (after 35.6 hours), and resulted in complete loss of control of pump pressure. It remained fixed at 8,000 psi regardless of input voltage. Abex found a broken filter screen on the EHV, which apparently led to contamination of the jet pump.

The second problem occurred with 112.2 hours on the pump and about 74 hours on the EHV, and was of a more mysterious nature. As system temperature increased, the pressure setting, with 10 volts applied, gradually climbed from 3,000 psi eventually reaching 8,000 psi when case drain fluid temperatures reached about 200°F. If the system was allowed to cool to room temperature level, the pump pressure could again be controlled over its whole range (3,000 to 8,000 psi). This was observed with both EHV's after 74 hours on the first and 26 hours on the second. Abex found a "nibbled" O ring on the sleeve of the first EHV, and it was returned to MCAIR.

Event No.	Date	Pump (hr)	Endurance (hr)	Stabilator Endurance Cycles	Event Description	Solution/Results
1	14 May 87	0	0	0	Started component acceptance/performance testing	
2	16 May 87	3.0	0	0	Excessive stabilator null leakage (5.8 gpm at 8,000 psi)	Returned load recovery valves to Berteau. larger stem in check valve required
3.0	16 May 87	3	0	0	Continued with pump performance tests, torque data inconsistent	
4	19 May 87	7.1	0	0	Replaced pump S/N 192174 with FWRHS pump S/N 192343 to verify torquemeter	Sent Acurex torquemeter to be calibrated to allow verification of MCAIR ring dynamometer unit
5	27 May 87	7.1	0	0	Continued with pump S/N 193174	
6	28 May 87	8.2	0	0	Began actuator baseline performance testing	
7	29 May 87	10.9	0	0	Adjusted distribution system to get proper no-load rate	7.8 in./sec; Spec: 7.5 + 0.8 in./sec
8	30 May 87	14.3	0	0	Actuator unstable at high temperatures	Adjusted valve gain
9	30 May 87	15.8	0	0	Took oil sample No. 1 at return filter sample port	Class 9 contamination level
10	06 Jun 87	17.1	0	0	Actuator unstable at all conditions	Used electrical gain compensator and lag filter to tune system
11	07 Jun 87	18.5	0	0	Completed leakage tests	0.85 gpm midstroke at 8,000 psi
12	11 Jun 87	21.3	0	0	Actuator unstable, force motor coils show 6 of 8 damaged	Force motor shipped to Berteau, replacement received on 12 Jun 87
13	12 Jun 87	22.8	0	0	Run pump horsepower tests, Torque data still inaccurate for MCAIR unit	Replaced hard lines to pump with hoses
14	15 Jun 87	27.1	0	0	High pressure hose failed, replaced with hard line	All future torque data recorded with Acurex torquemeter
15	16 Jun 87	27.9	0	0	Pressure filter boss seal failure	Nibbled O rings replaced
16	22 Jun 87	34.9	0	0	During 100% stall load tests, transfer tube buckled	Berteau sends replacement
17	23 Jun 87	35.6	0	0	Continued with heat rejection tests, pump EHV failed after 35.55 hr	Replaced with backup pump EHV, Abex sends replacement
18	25 Jun 87	37.0	0	0	Pump case horsepower/drain anomalies	Installed backup pump S/N 193175
19	25 Jun 87	37.0	0	0	Horsepower data more predictable	Re-install pump S/N 193174
20	29 Jun 87	38.1	0	0	Continued horsepower/case drain anomalies	Shipped S/N 193174 to Abex Re-installed pump S/N 193175
21	13 Jul 87	38.1	0	0	Completed pump heat rejection tests, transfer tube arrived from Berteau	
22	15 Jul 87	38.1	0	0	Completed baseline actuator testing, installed 0.003 in overlapped valve	
23	23 Jul 87	38.1	0	0	Actuator showing signs of instability, poor hysteresis, chatter, drift	Used electrical gain compensator to minimize effects
24	27 Jul 87	38	0	0	LVDT housing seal failure during 40% loaded frequency response	Replaced broken O ring
25	12 Aug 87	38.1	0	0	Transfer tube buckled during loaded hysteresis test	Berteau sends replacement on 14 Aug 87

Figure 344
Chronological Summary of Testing

Event No.	Date	Pump (hr)	Endurance (hr)	Stabilator Endurance Cycles	Event Description	Solution/Results
26	20 Aug 87	38.1	0	0	Actuator performance problems (per #23) still present, valve teardown revealed failed MCV piston seals	Returned entire actuator to Berteau for repair and test
27	20 Aug 87	38.1	0	0	Oil sample No. 2 taken downstream of return filter	Class 6 contamination level
28	24 Aug 87	38.1	0	0	Abex returned pump S/N 193174	Re-installed pump S/N 193174, 47.9 hours on pump S/N 193174
29	14 Oct 87	39.2	0	0	Berteau returned actuator with new overlapped spool	
30	16 Oct 87	39.5	0	0	LVDT housing seal failed, LVDT cores bent, bearing failed	Replacement parts sent from Berteau, poor installation blamed for problems
31	21 Oct 87	39.5	0	0	New LVDT cores arrived	Performance testing resumed on 23 Oct 87
32	22 Oct 87	39.5	0	0	Oil sample No. 3 taken downstream of return filter	Class 4 contamination level, water content: 181 ppm
32	28 Oct 87	42.5	0	0	Pump outlet check valve failed	Replaced check valve
33	02 Oct 87	58.6	0	0	Completed overlapped valve performance testing, added load recovery valves to manifold	
34	05 Nov 87	61.3	0	0	Completed load recovery valve performance testing, added jet pumps to manifold, readjusted distribution system line drops	
35	16 Nov 87	72.9	0	0	LVDT housing seal failed	Replaced, with backup ring, added
36	10 Dec 87	77.2	0	0	Inner groove cap strip on MCV failed	Replaced cap strip
37	11 Dec 87	82.2	0	0	Actuator performance deteriorating	No problems found
38	14 Dec 87	85.2	0	0	Oil sample No. 4 taken downstream of oil filter	Class 5 contamination level
39	14 Dec 87	86.1	0	0	Began endurance test using constant 8,000 psi pressure (to allow time for development of start pump control box)	
40	16 Dec 87	90.1	3.8	19,000	LVDT housing cracked (bad weld on probe tube)	Returned failed unit to Berteau, replacement arrived 18 Dec 87
41	04 Jan 88	95.5	3.8	19,000	Began variable pressure pump operation	
42	05 Jan 88	98.5	6.8	34,000	Oil sample No. 5 taken downstream of return filter	Class 10 contamination level, water content: 113 ppm
43	05 Jan 88	101.3	9.0	45,000	Stopped endurance to run performance test, readjusted distribution system to reduce speed differences due to direction	
44	07 Jan 88	106.8	9.0	45,000	Completed jet pump retest, restart endurance test	
45	07 Jan 88	112.2	13.0	65,000	Lost control of pump variable pressure	Replaced EHV with backup, shipped to Abex for analysis
46	08 Jan 88	112.2	13.0	65,000	Oil sample No. 6 taken downstream of return filter, torn element discovered, filter seals replaced	Class 12 contamination level, water content: 142 ppm, return filter element replaced
47	13 Jan 88	139.3	39.4	197,000	Lost control of pump pressure as fluid temperature rose above 110°F	Continued running at constant 8,000 psi

Figure 344 (Continued)
Chronological Summary of Testing

Event No.	Date	Pump (hr)	Endurance (hr)	Stabilator Endurance Cycles	Event Description	Solution/Results
48	14 Jan 88	145.0	44.6	223,000	Lost control of pump pressure as temperature rose above 110°F	Continued endurance test at constant 8,000 psi
49	21 Jan 88	149.4	48.4	240,500	Weld on actuator inertia wheel failed	Repairs made by 19 Feb 88
50	17 Feb 88	149.4	48.1	240,500	First EHV is returned by Abex with new seals, checkout test run	Backup EHV shipped to Abex for inspection and repair
51	26 Feb 88	149.7	48.1	240,500	Actuator MCV LVDT began leaking at the same place as the previous unit	Berrea sends backup, add backup plate to leaking LVDT
52	08 Mar 88	153.0	48.5	242,500	Oil sample No. 7 taken downstream of return filter	Class 3 contamination level
53	08 Mar 88	155.5	49.0	245,000	Oil sample No. 8 taken downstream of return filter	Class 2 contamination level
54	11 Mar 88	190.9	84.4	422,000	Low reservoir level shuts down system automatically	Replaced leaking needle valve
55	14 Mar 88	204.7	98.0	490,000	Variable pressure only going down to 4,000 psi, load system acted peculiar	Continued running
56	14 Mar 88	211.6	103.6	518,000	Checked actuator null leakage and pump case drain flow	Valve null leakage: 0.42 gpm, Pump case drain flow: 0.93 gpm.
57	14 Mar 88	211.8	103.8	519,000	Oil sample No. 9 taken downstream of return filter	Class 2 contamination level
58	15 Mar 88	229.8	121.8	609,000	Load cylinder rod bearing failed	Continued running unloaded
59	15 Mar 88	232.9	124.9	624,500	Pump pressure stuck at 8,000 psi	Continued running, still waiting for backup EHV to return from Abex
60	16 Mar 88	243.0	139.4	697,000	Actuator chattered badly, Disassembled main control valve	Installed new LVDT, replaced several O rings, checked force motor coils, also tried line-to-line spool
61	25 Mar 88	245.0	139	697,000	Problem still unsolved, MCV now dithered at 40 Hz for an undetermined reason	Shutdown testing

**Figure 344 (Concluded)
Chronological Summary of Testing**

The third failure occurred 125 hours into the endurance test (232.9 total pump hours). The pressure stuck at 8,000 psi, even after a 10 volt command was applied. Endurance testing continued until the backup EHV was returned from Abex.

Another anomaly was discovered with the pump. Repeatedly, case drain flows were recorded as being higher at 3,000 and 5,500 psi than at 8,000 psi. This occurred on both 15 gpm smart pumps consistently. After the first EHV failure, the pump was sent back to Abex for investigation of this phenomenon. With the EHV electrically controlling the pump compensator setting, Abex also reported higher internal leakage at lower pressure settings. However, when the pump compensator was mechanically locked at a specific pressure, case drain flows returned to a more normal trend (i.e., higher at 8,000 psi). Therefore, Abex concluded that the higher leakage was caused by the EHV return control flow which was dumped into the pump case drain, and that there was no problem with pump mechanics.

(2) Stabilator Servoactuator Experience - A number of problems were discovered with the Bertea stabilator actuator before the endurance test was run. The first test performed on the actuator was a null leakage test, and the results were much higher than expected (5.8 gpm). The load recovery valves were replaced with plugs and shipped back to Bertea for evaluation. The stem of the plug had been designed too small, and allowed the plug to tilt from side to side under pressure. Four new plugs were made and shipped to MCAIR bringing the leakage to normal levels.

While attempting to run leakage tests at 275°F, pump case drain fluid temperature, the actuator could not be held steady. Oscillations developed in the main ram and could not be controlled. The electronics were checked by lab technicians, however, no problems were discovered. Initial frequency response testing revealed a peak around 20 Hz, which corresponded closely with the natural frequency of the inertia wheel used to load the actuator (23 Hz). Using an in-house electrical gain compensator and a lag filter to tune the system, control of the actuator was eventually achieved by putting a lag in the forward loop and cutting the feedback loop gain. Interestingly enough, after switching to the overlapped MCV, the gain compensator was no longer required.

While running valve reversals, the actuator suddenly experienced another bout of uncontrollability. The coils on the DDV force motor were checked, and it was discovered that six of the eight half coils were inoperable. A replacement was shipped from Bertea soon after, and testing resumed.

The next significant failure was the transfer tube between the valve and the fourth cylinder chamber, shown in Figure 345. This was the first of three tubes to buckle under an 8,000 psi, fully loaded step. One of the casualties included a thicker walled tube that was supposed to solve the problem. For the sake of our test setup, the problem was temporarily solved by clamping a large metal sheath, which completely enclosed the tube, to brace it. A permanent solution to this problem requires a straight balanced tube or straight and bent tubes that are well retained at each end.

A chronic problem with the LVDT housing seal developed (shown as "1" in Figure 346). A large (-22) O ring which sealed the LVDT housing, ruptured with as little as two hours of testing. Replacement of this seal as well as the inner groove seal ("2" on Figure 346) of the end cap, which allowed flow

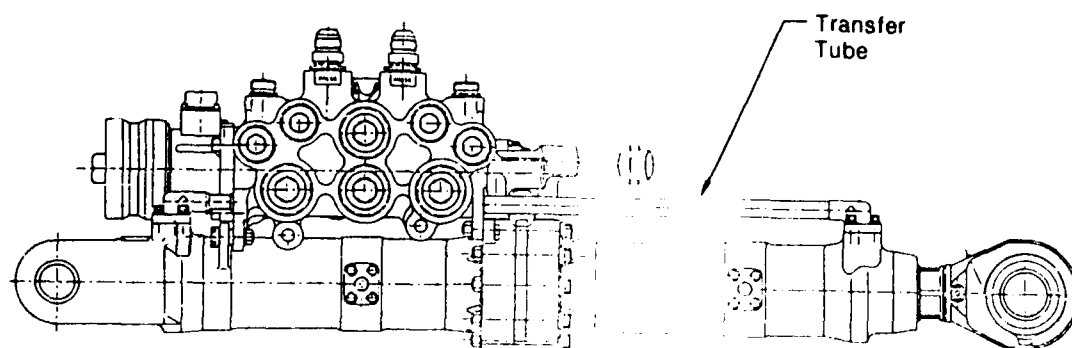


Figure 345
Schematic of Actuator Transfer Tube

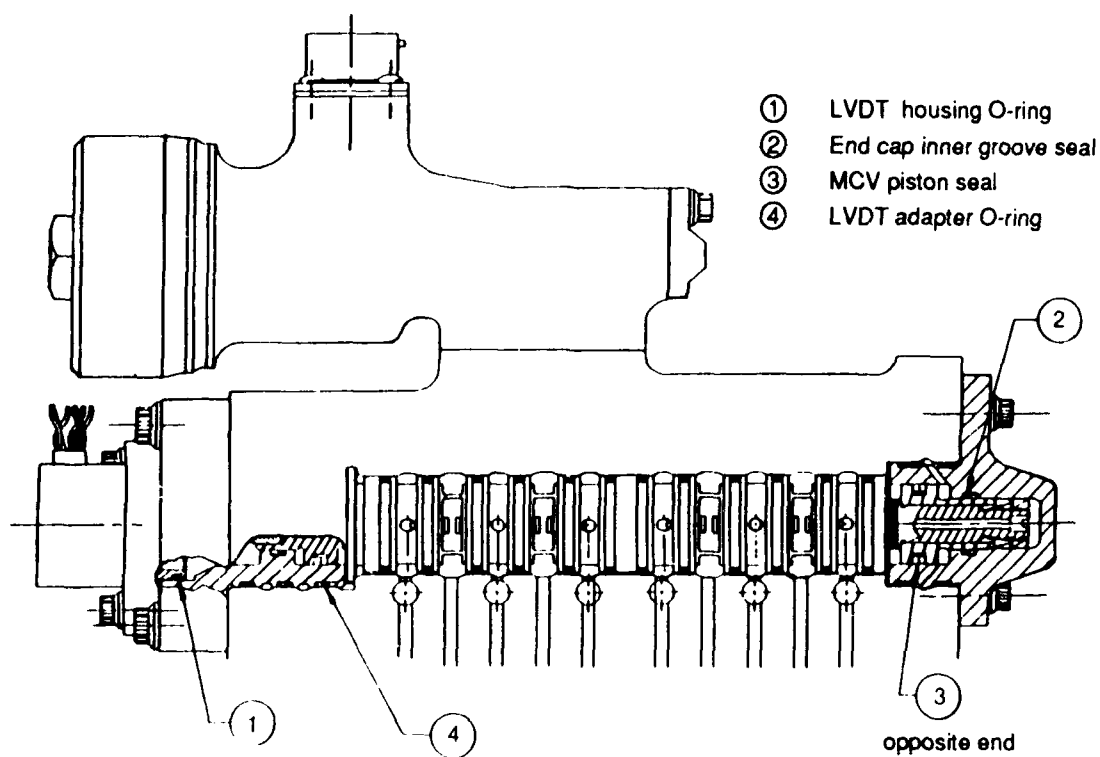


Figure 346
Location of Seal Failures on Stabilator
Actuator Main Control Valve

by way of the hollow spool to reach the LVDT housing seal, helped for awhile. The problem was finally solved by fabricating a backup ring for the groove.

After completion of performance testing on the actuator with a line-to-line valve, the spool was removed and replaced with the 0.003-inch overlapped version. Initial performance tests carried out showed poor performance, with significant chatter present on retract strokes, and a stall load that was too low. Hysteresis tests performed on the actuator, revealed a major performance problem. On the extend side, the hysteresis was about 1.5 inches, while there was zero on the retract side. In addition, the valve would occasionally get stuck in the hardover extend position, causing erratic actuator behavior. The valve was torn down, and it was discovered that the MCV modulating piston seals were broken. The broken seals (shown as "3" in Figure 346), apparently had caused the valve to stick. In addition, the modulating piston sleeve had been scraped by the energizing ring.

The reason for the seal problem was readily apparent. One of the two seal retaining balls on the energizing ring was missing (Figures 347 and 348). The two balls are used to keep the seal cuts from lining up, which would result in higher than normal valve leakage. According to Parker Berteau, it is likely that the seals were damaged on installation of the overlapped control valve spool. Once it was discovered how to properly install the MCV, no further problems were encountered with the modulating piston seals.

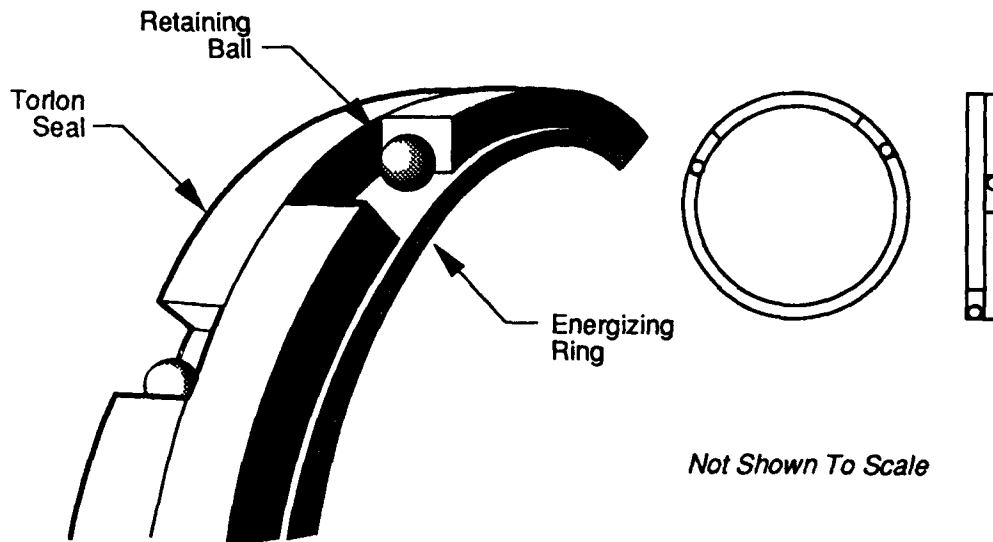


Figure 347
Actuator Main Control Valve Modulating Piston Seals

MAIN CONTROL VALVE PISTON SEAL



Figure 348
Photo of Broken Seals

Further performance problems with the actuator occurred while running flow augmentation tests. Examination of the MCV LVDT core assembly, revealed that it had been damaged, apparently on installation. Several balls were missing in the LVDT ball bearing, resulting in a considerable amount of axial travel, and all four LVDT probes were noticeably bent. A new LVDT bearing and four new probes were sent from Bertea and installed at MCAIR.

After less than 5 hours of endurance testing (plus some 165 hours of performance testing), the MCV LVDT housing began leaking severely. A welded joint between the coil form and the base cracked, and allowed fluid to escape past the "can" which enclosed the four coils. Three of the four welds retaining the can had cracked due to the leak. The LVDT was replaced with an F-18 type and testing resumed.

The biggest problem encountered during endurance testing occurred at 138.7 hours (238.6 total pump hours). The actuator began to chatter when retracting. Dismantling the valve showed no unusual wear, although a few O rings were bad (see "4" in Figure 346). The O rings were replaced and a new LVDT was mounted. The main ram no longer chattered, but the MCV was now dithering at about 40 Hz. No control problems were evident, so the problem appeared to be force motor related.

(3) General System Experience - Few problems were encountered with the remaining systems and components. A description of the major problems follows.

Torquemeter - To measure torque, it was initially planned to use a MCAIR designed ring dynamometer style torquemeter. The design used a 100-lb capacity load cell mounted on an arm 8.5 inches from the center line of the unit. The load cell signal passed through the Neff data collector and was calibrated to read out in units of torque (in-lbs). However, initial testing determined that there was a significant error induced by the pressure, case drain and suction lines connected to the pump. These lines were replaced temporarily with flexible hose, and more consistent data resulted. An Acurex Autodata 1200A torsion measurement system was mounted simultaneously as a comparison. The Acurex used strain gages mounted directly to the pump input shaft. The signal from the strain gages was transmitted to an antenna that circled the shaft. The signals were converted into torque and recorded. Since this design measured shaft torque rather than housing torque, it was not affected by external factors such as lines or bearing friction.

Although the ring dynamometer torquemeter showed the same results as the Acurex when hoses were used, a ruptured high pressure hose necessitated a switch back to titanium tubing, which greatly reduced the effectiveness of this instrument. For all LECHT energy comparisons, the Acurex torquemeter data was used.

Actuator Inertia Wheel and Load Cylinder - During the endurance test, the weld which held the shaft to the inertia wheel failed. An examination revealed a poor weld. This inertia wheel had been through approximately one thousand hours of actuator testing, having been utilized for the FWFRHS test as well as LECHT. A new shaft was welded to the inertia wheel, and new bearings replaced the old ones which had been damaged in the failure. In addition, a new bushing for the actuator horn was installed. After approximately 60 hours, the endurance test was again shut down due to a failed load cylinder rod end bearing. This ended the loaded portion of the testing.

5.3 TASK 3 - HYDRAULIC FLUID SAMPLES

5.3.1 Approach - During the performance and endurance testing, samples of CTFE fluid were taken from a sampling port downstream of the system return filter. Half of each sample was sent to AFWAL/MLBT for analysis and half was sent to the MCAIR Analytical Chemistry Laboratory (ACL).

a. MCAIR Analytical Chemistry Laboratories Tests - The samples received by MCAIR ACL were initially contamination patch tested, using a 100 milliliter sample filtered through a 0.45-millipore laboratory filter apparatus. The filter paper with residue was flushed with filtered petroleum ether to remove the CTFE fluid, then vacuum dried. A Bausch and Lomb microscope with a 40 to 1 magnification eyepiece micrometer disc, was used in counting the particles. Samples of any particulate were removed and mixed with a potassium bromide (KBr) pellet, fused together under glass and pressure, then analyzed with a Nicolet Instrument Corporation Model NIC-170

Fourier Transform Infrared Spectrophotometer. This instrument gives a wave number trace of the substance. It then reproduces a wave number trace of a known compound from its computer memory to compare with the sample. Nothing out of the ordinary was discovered with the fluid samples.

b. Fluid Analysis Result - The MCAIR results of the fluid samples taken are summarized in Figures 349 and 350. The emphasis of the MCAIR fluid sample testing was on establishing fluid contamination levels. Figure 349 shows that the contamination level varied continuously throughout testing, ranging from Class 2 to Class 12.

At the beginning of the endurance test, the contamination level of the fluid was very high (Class 10 and 12). After taking the sixth fluid sample, it was discovered that the return filter element had been torn, and was extremely contaminated. Among the items found in the torn element were the remnants of numerous aluminum flare-savers that were being used throughout the system on the AN type fittings. Following the replacement of the element, only steel flare-savers were used in the system. It should be noted that during this period of high fluid contamination, the pump EHV failed several times. Although the reasons for the EHV failure were never specifically mentioned as contamination by Abex, EHV's are very sensitive to fluid contamination. Once the torn element was replaced, the fluid samples that followed were relatively clean (Class 3 or less).

It should be noted that the analysis of fluid contamination is not an exact science, and there is some variation due to personnel changes. In addition, the fluid contamination levels can be thrown off by outside contaminants, either in the air or in the sample container.

On four of the nine samples taken, water content was recorded as well, and varied between 113 and 208.7 ppm. Acidity varied between 0.125 and 0.230 Mg-KOH/gm.

The Airforce AFWAL/MLBT further evaluated similar fluid samples by gas chromatography and infrared spectroscopy. No significant differences were observed by these techniques. The fluid appeared to have been stressed very little as indicated by the low ACID number and light color as reported in Figure 351. On one sample, a four ball wear test was also performed resulting in a wear scar of 0.65 mm. All of these results are similar to those of unstressed fluid.

SECTION VI CONCLUSIONS AND RECOMMENDATIONS

6.1 CONCLUSIONS

The completion of the demonstration tests and 139.4 hours of the endurance test allowed several conclusions to be drawn about the four low energy concepts and the various components used to demonstrate them. Figure 351 is a summary of the four concepts and the conclusions drawn from testing.

Testing Confirmed Concepts Predicted Capabilities

- Overlapped Valves
 - Acceptable Dynamic Performance on Simulator
 - Null Leakage Reduced to 20% of Line-to-Line
 - Eliminated Digital Dither Null Flow
- Variable Pressure Pump
 - Heat Rejection: 3,000 psi (Low Setting) Is 50% of 8,000 psi (High Setting)
 - Variable Pressure System Verified on Simulator
- Flow Augmentation/Load Recovery Valves
 - Cuts Peak Flow Demand in Half on F-15 or F-18
 - Acceptable Performance on Simulator
 - LRV Peak Rate Three Times Conventional
 - Maximum Application Could Give Added 5 - 10% Wt Savings
- Full Up Variable Pressure System
 - "75% at 3,000" Duty Cycle Reduces System Energy Requirements by Half

Figure 352
LECHT Program Results

Overlapped Valves - Valve overlap allowed 62 to 78 percent reduction in steady-state null leakage and associated heat rejection at the actuator. In a system that consisted of several actuators, this savings was quite significant. A side benefit of overlapped valves was the elimination of digital computer induced noise. On the F-18, it was discovered that digital noise was doubling the null leakage on the flight control actuators.

Some performance degradation was noticed on small amplitude frequency response tests. However, previous F-15 iron bird and F-18 simulator evaluations showed that aircraft handling qualities were acceptable, if not superior, with overlap valves.

The conclusion on overlapped valves is that a significant reduction in system heat due to valve leakage can be achieved.

Variable Pressure - Tests indicated that at steady-state, no-flow conditions, a 45 percent reduction in pump heat rejection was achieved at 3,000 psi vs. 8,000 psi. Varying the system pressure between 3,000 psi (75 percent of the time) and 8,000 psi (25 percent of the time) depending upon system demand, allowed a 48 percent reduction in energy consumption vs. a constant 8,000 psi system over the endurance test duty cycle. An additional benefit of variable pressure operation, is that the reduced heat generated allowed a smaller system heat exchanger thus saving weight. Also, resulting longer component life reduced life cycle costs.

The conclusion is that a significant reduction in system heat rejection, and therefore energy, can be realized with a properly designed, variable pressure system.

Flow Augmentation - The jet pump utilized the relatively high return pressure and flow leaving a servoactuator during high surface rate conditions to supplement the inlet flow to the actuator. The advantage to using a jet pump flow augmentor, was a 32 to 56 percent reduction in central system flow demand which would allow the use of a smaller pump, filter, reservoir, etc. A smaller heat exchanger could be used to take advantage of the smaller pumps reduced heat rejection. A potential 5 to 10 percent additional system weight savings existed.

The conclusion is that for an actuator requirement compatible with flow augmentation capabilities, flow augmentation can significantly reduce central system peak flow demand and thus system weight. For actuators requiring high rates at high loads, the use of rate limiting by the flight control computer can probably achieve better results.

Load Recovery Valves - The use of load recovery valves allowed a significant increase in actuator rate (by almost 300 percent), during periods of high assisting loads. There was a potential for reduced central system peak flow requirements where the resulting higher average rate allowed a reduction in peak actuator flow demand.

6.2 RECOMMENDATIONS

The following recommendations were derived from the LECHT program. Some of the lessons learned on the LECHT program have been incorporated into the Nonflammable Hydraulic Power Systems for Tactical Aircraft (NHPSTA) program, and other 8,000 psi CTFE research efforts.

Transfer tube design - The numerous failures that occurred on the transfer tube before it was finally braced, indicated a bad design. The use of unbalanced transfer tubes with high L/D ratios at 8,000 psi, should only be allowed if both ends are securely fastened. A balanced tube design would probably be a better alternative.

LVDT design - All LVDTs exposed to return fluid pressure, should be designed to at least 5,000 psi operating pressure and 8,000 psi burst pressure for an 8,000 psi actuator. Return system pressure transients of 4,800 psi were not unusual, and repeated applications of these cycles caused two different main control valve LVDTs to fail. Both had been designed to a 1,500 psi operating return pressure associated with their 3,000 psi design requirement.

Low Energy Concepts - All four concepts demonstrated in the program have been adapted for the NHPSTA program, and will allow a look at the concepts on a aircraft system level.

SECTION VII
REFERENCES

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2. Report, NADC 79024-60, "Fabrication and Testing of Lightweight Hydraulic System Simulator Hardware - Phase 2," January 1986.